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GUIDANCE SYSTEMS
DIVISION

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ATM CMG
BEARING
FAILURE ANALYSIS
FINAL REPORT

JANUARY 1975

PREPARED FOR


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ALABAMA

6401-75-R06


S. SEEB
DIRECTOR OF ENGINEERING


E. C. REUTTER
DIRECTOR OF MARKETING


B. J. O'CONNOR
GENERAL MANAGER

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SECTION 1.0

INTRODUCTION

The primary mechanisms used to orient and maneuver the Skylab Space Laboratory were three Control Moment Gyros (CMGs) manufactured at Bendix Guidance Systems Division.

Each CMG used two 107 H size angular contact ball bearings to support a 150 pound wheel rotating at a speed of 9100 rpm and developing an angular momentum of 2300 ft-lb-sec. Each of the gyros were tested at Bendix GSD and at NASA, Huntsville, Alabama, for approximately 1500 hours prior to their launch in the Skylab Space Laboratory on May 14, 1973.

On November 23, 1973, after 195 days of continuous operation in space, ATM CMG S/N 5 (Skylab #1) ceased to function due to a probable bearing failure. During the remaining 75 days of the mission, close monitoring of speed, current and temperature data from the remaining two functioning ATM CMGs revealed periods of slightly unstable behavior in ATM CMG S/N 6 (Skylab #2).

The unstable behavior manifested itself by slight speed, current, and bearing temperature fluctuations about the norms but did not affect the functioning of the gyro. However, the trend of the data indicated a possible deterioration in the performance of one of the two bearings paired in this gyro.

Both of these remaining ATM CMGs completed the mission of approximately 6500 hours in space and were run down

after the egress of the Skylab IV astronauts.

Task Order 25-74 of Contract No. NAS 8-20661 was issued to the Bendix Corporation Guidance Systems Division to study the ATM CMG data from Skylab, to initiate a complete design review and to assess the cause or causes for the failure of ATM CMG S/N 5 and for the anomalies experienced with ATM CMG S/N 6. The enclosed reports address (see Appendices) each of the areas investigated under the task order.

SECTION 2.0

DISCUSSION

The investigation into the cause or causes for the failure of ATM CMG S/N 5 (Skylab #1) and the anomalies associated with ATM CMG S/N 6 (Skylab #2) was divided into the following four areas:

1. Review of the Skylab data
2. Investigations into specific areas by outside consultants and a literature search
3. Bendix tests and simulations
4. Bendix design reviews

The review of the Skylab data was divided into eleven periods. These periods were selected to try to isolate and/or correlate various factors possibly related to the failure and the anomalies observed. The periods were selected to determine long term trends in wheel speed, currents and temperatures and how these parameters were related to various maneuvers, temperature cycles, gravity gradient desaturations, docking and undocking and periods of manned and unmanned operations.

The primary source for the data reviewed was the Skylab telemetry data. Other data sources were also used where necessary. A computer program was utilized to present this data in the form of parameter distributions. The report on the details of the data review is presented in Appendix A.

Because of the size of the report generated during this task, we have not been able to include it within this volume. This data review report and its appendix is on file at NASA and is available for review there.

The investigations by outside consultants were performed to develop more fully specific areas of interest, and to reinforce the initial theory that the problems associated with ATM CMGs S/N 5 and S/N 6 were caused primarily by marginal bearing lubrication.

Other areas of investigation included the possible occurrence of bearing ball speed variation (BSV), the conditions necessary for its initiation, and the effects if any. Also investigated was the effect of centrifugal forces on the lubricant present on the bearing rotating inner race and bearing balls. The effects of thermal gradients and surface conditions (specifically the presence of retainer transfer material) on the migration of oil from the bearing outer race land to the contact area was investigated. The dynamics of retainer instability, particularly under the conditions of high lubricant viscosity and marginal lubrication, were also studied. The reports submitted by the consultants and a summary of their efforts appear in Appendix B.

A literature search was conducted into phenomena associated with the affects of orbital conditions, particularly hard vacuum and zero "g", on lubricants. A total

of forty-five documents were reviewed some of which were helpful in further understanding certain areas relevant to the ATM CMG situation on Skylab. The results of this search appears in MT 15,785 in Appendix BV.

The Bendix testing was concerned primarily with various bearing tests to investigate the effect of lubricant or lack of lubricant in the ATM CMG bearings and the dispersion and migration of the lubricant. The tests were all conducted to simulate, to a certain extent, the vacuum and weightless conditions of space. Four life tests were conducted with bearings containing a redesigned bearing retainer for the purpose of proving the effectiveness of the design change. The reports on these tests may be found in Appendix C VII and C VIII. In addition, three tests were conducted in which minimally lubricated bearings were monitored and run to imminent failure. The purpose of these tests was to attempt to simulate and correlate the bearing conditions and the recorded variations in speed, current and temperature with the data telemetered from the Skylab ATM CMGs. The report on these tests is presented in Appendix CIII.

To determine the degree of dispersion and migration of oil in a bearing with standard retainers, a test was conducted utilizing a minimally lubricated bearing with its spin axis oriented such that the migration of any resupplied oil would be against lg. The test was conducted using red dyed oil in the resupply system such that the dispersion and migration of the oil was readily visible. It is assumed that if the resupplied oil does

migrate or is pumped into the race against lg, then an equal or greater amount would similarly enter the race in a 0g space environment. The report of this test appears on Appendix CV. Appendix CIV contains a report of a test to determine the life expectancy of an ATM CMG bearing which has sufficient lubrication initially but is not resupplied with lubricant.

Other tests were conducted to determine the effect of temperature, particularly lower temperatures, on lube nut flow rate and ATM CMG bearing torque. The reports covering these tests appear in Appendices CI and CII, respectively. Appendix CVI contains a report on bearing lubricant hydrodynamic film tests.

The Bendix design reviews included investigation into areas most of which had been reviewed previously. These areas included the lube nut design and flow rates, bearing stresses and loads, wheel stresses, electric motor design, and bearing fits and the bearing retainer. The fact that the retainer feed holes were positioned to direct the supply oil to the land adjacent to the race way rather than directly to the ball-race way control angle was previously noted and was not considered to be a problem. However, upon further investigation and review, Bendix GSD now considers this to be the major failure mechanism. A modified retainer was then investigated and has corrected this condition.

SECTION 3.0

RESULTS AND CONCLUSIONS

Analyses of the results of the tests conducted in this task and the consensus of opinion of the consultants commissioned, points to inadequate lubrication as the predominant factor causing the failure of ATM CMG S/N 5 (Skylab CMG #1) and the anomalies associated with ATM CMG S/N 6 (Skylab CMG #2).

The problem of inadequate lubrication was caused by the inability of the makeup supply oil to migrate from the outer race land to the immediate bearing contact area. This was caused by the original positioning of the make-up oil on the land and the oil wetting restriction which inhibited migration. This migration restriction was due to thermal gradient effects and possibly retainer debris and oil polymerization after the failure mechanism was established and the problem was compounded.

Retainer loads resulting from Ball Speed Variation during maneuvers could have added to the retainer debris for an inadequately lubricated bearing.

This problem did not manifest itself during exhaustive testing on earth. In five years of testing prior to launch, a total of 480,000 hours were accumulated on 22 units including bearing tests in life test fixtures and wheel qualification tests. We believe the failure mechanism was masked by the pooling of oil in a 1 g field which would supply oil whenever the spin axis was horizontal. Although it now appears possible that the failure of ATM CMG S/N 11 during qual tests was precipitated by the same mechanisms, (the cause assignment was low rate of flow from the lube replacement system) it was not designated such by the investigative

team of Bendix, NASA, and various consultants and contractors at the time of its failure.

As a means of showing that the original Skylab retainer did not supply sufficient lube, a series of tests were run (MTs 15,781, 15,784 and 15,786) with marginally lubed bearings in the SA vertical position. In all cases, with the holes supplying oil to the lands (as in the Skylab CMGs), we were able to induce failure in less than 200 hours. Since the design modification was made we have been able to run marginally lubricated bearings for thousands of hours. These tests with both new and old retainers were performed several times and always yielded the same results (the old retainers failed and the new ones survived).

As a result of all of the efforts discussed, Bendix GSD believes that the failure mechanism associated with the bearing retainer holes caused the problem in the Skylab CMG and was probably slowly starving the second CMG. If allowed to run further, we also believe that the third CMG would have eventually suffered problems when it depleted the oil supply on its retainer. It is also felt that the P-11 failure in Houston with the S.A. vertical can also be attributed to this same failure mechanism.

Further testing of the new and old retainer designs have been proposed to further validate the above premise.

Other than retainer hole relocation, the design reviews and studies did not disclose any areas where further design changes are necessary.

There is nothing in the data search that implies that external events were responsible for the failure of Skylab CMG #1 and the anomalies associated with Skylab CMG #2.

From the data reviewed it was observed that normal changes in speed, current and bearing temperature did occur with coincident bearing heater cycles, periods of high solar intensity, and with changes in electronics assembly temperature. There were also changes related to docked and undocked modes. There were small changes in bearing Δ temperature related to bearing heater cycle and change in EA temperature. However, the primary sign denoting anomalies in the bearings of CMG S/N5 and CMG S/N 6 and the impending failure of CMG S/N 5 were changes in the bearing Δ temperature in excess of normal changes due to the above effects. It also appears that bearing number one (the failed bearing) in CMG S/N 5 always had slightly higher heat generation than its opposite number. This temperature generation appears to trend upward with time particularly during the weeks prior to its failure. There is no such consistent trend in Δ temperature associated with the bearings on CMG S/N 6 or CMG S/N 7.

The gravity gradient desaturations and other maneuvers do not appear to affect the CMG wheel speeds or currents

since the expected small changes are within the granularity of the telemetry equipment. However, there appears to be some induced gyro thermal variations related to some of the CMG S/N 6 anomalies. Although the effect of the gravity gradient dumps were not noted during the earlier phases of the mission, there was some correlation noted during the later phases between the anomalies and the occurrence of "g-g" desaturations.

From the results of MT 15,774 (Appendix CI) the flow of the lube nuts in the Skylab CMGs was sufficient to adequately relubricate the bearings. The results of the thermal vacuum test (MT 15,783, Appendix CII) indicated an expected change in bearing torque of approximately 25-33% in the Skylab CMG bearing temperature cycle range of 60°F to 80°F. This was due to the change in viscosity of the KG 80 oil used to lubricate the bearings and accounts for the changes in speed and current associated with the temperature cycle and periods of high solar intensity.

The results of MT 15,775 indicate that an ATM-CMG Bearing would run for at least 1500 hours and probably in excess of 3000 hours without additional lubricant other than the lubricant in the ball bearing retainer and the residual oil on the bearing races and the balls.

The results of tests to determine whether the resupplied oil would migrate against lg is documented in MT 15,776 and indicates some oil does migrate or is pumped against lg into the bearing outer race. However, most of the

oil does not and the quantity and rate of migration into the race is insufficient for adequate relubrication. However, this would support the supposition that resupplied oil could reach the bearing race from the land in a "0" g space environment. However, the minimal amount moving this way against the thermal gradient flow was probably not enough to sustain bearing life.

The results of MT 15,782, an investigation to determine the adequacy of the bearing lube hydrodynamic film indicated that the speed at which the hydrodynamic film is developed in a normally lubricated bearing is 66-84 rpm. Therefore, an adequate film is available if the bearing is adequately supplied with oil.

The results of life tests utilizing bearings with redesigned retainers (MT 15,781, MT 15,784) indicate that the modified retainer redesign described in MT 15,777 does deliver all of the resupplied oil directly to the bearing outer race and at a rate that is more than adequate to properly relubricate the bearing.

The results of tests which attempted to simulate the failure of CMG S/N 1 and the anomalies of CMG S/N 2 (MT-15,786) by running marginally lubricated bearings in a life test fixture were vague. Although some similarities appear, both for the failure of CMG S/N 1 and the "glitches" of CMG S/N 2, overall correlation between minimal lubrication and the Skylab anomalies was not accomplished.

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The fact that all of the CMG's lasted at least 195 days (4680 hours) agrees with preliminary earth tests that predicted at least 1500 to 3000 hours of running with no supplemental lubrication. The additional time beyond the 3000 hours can probably be attributed to the reduction in load associated with zero G operation (i.e. removal of the 150 lb. wheel weight).

Concurrent with the writing of this report GSD has been awarded a follow-on contract to further validate our failure mechanism premise. As part of this new program we will continue running our three life test fixtures and unit E-2 (all with the modified retainer design). We will also do further testing to compare the old and new retainer designs by utilizing two more life test fixtures and the IGRA of Unit S/N 4.

SKYLAB ATM/CMG DATA REVIEW

DUE TO THE SIZE OF THE ATM-CMG DATA REVIEW
AND ITS APPENDIX, IT IS NOT INCLUDED IN
THIS VOLUME. THIS REPORT AND ITS APPENDIX
IS ON FILE AT NASA, HUNTSVILLE AND IS AVAIL-
ABLE FOR REVIEW THERE.

To: Engineering File MT-15,787
From: T. Wheelock

Issue: Original
Date: 10 January 1975

ATM CMG BEARING FAILURE

CONSULTANT REPORT

Prepared by:


T. Wheelock

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608

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1.0 DEFINITION OF CONSULTANT EFFORT

The nature of the failure of Skylab CMG #1 (P-5) and the anomalies of CMG #2 (P-6) had been discussed and analyzed by several organizations: (Draper Lab, Battelle); NASA facilities; (MSFC, Lewis Research Center, Langley Research Center) and Industrial organizations; (Bendix, Barden Corp., IBM). Some preliminary agreement as to probable failure modes had been established. Certain aspects of the problem had not been fully developed and it was agreed the other sources of bearing technology should be contacted for more complete analysis of these areas.

These consultant organizations and individuals, to be subcontracted through BGSD included Battelle (Columbus Lab.,) Tom Barish (consulting engineer) and A. B. (Bert) Jones (consulting engineer). These three consultants had been involved with an earlier CMG (P-11) failure analysis. A basic program outlining the areas of investigation for each consultant was proposed. (See Table 1).

TABLE 1

B. Consultant Support

B 1 Battelle

- 1.1 Lube migration phenomenon in 0 g and hard vacuum.
 - 1.1.1 Surface tension - surface energy equilibrium at lube vapor pressure.
 - 1.1.2 Surface adhesion.
 - 1.1.3 Circulation and flow in 0 g's.
 - 1.1.4 Effect of surface temperature gradients.
 - 1.1.4.1 Lube flow.
 - 1.1.4.2 Lube evaporation.
 - 1.1.5 Long term effects at hard vacuum and radiation environment.
- 1.2 Ball speed variation.
 - 1.2.1 Expansion of preliminary report critical range etc.
 - 1.2.2 Transient loading. Torque pulses.
- 1.3 Labyrinth seal analysis.
 - 1.3.1 Analysis of current labyrinth seal and housing orifice design using temperature and pressure data profile.
 - 1.3.2 Theoretical analysis 0 g effects (if any).

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TABLE 1 (Continued)

- 1.4 Retainer Instability
 - 1.4 General testing, possibly with unstable retainers, continuous monitors, power, torque temperature vibration, etc.
 - 1.5 Tests of outer race control bearings - (LTF) may be done at BGSD.
- B 2. A. B. Jones
- 2.1 Ball speed variation and computer analysis for 0 g conditions.
 - 2.2 Computer run - orbit.
 - 2.2.1 Orbit steady state.
 - 2.2.2 Orbit Torqueing.
 - 2.2.3 Orbit transient torqueing.
- B 3. Tom Barish
- 2.1 BSV analysis, orbit conditions.
 - 2.2 Transient torque.
 - 2.3 Normal torqueing in orbit.
 - 2.4 Possible experience with lube in space environment.

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Although several phone conversations with Bert Jones were made it was determined that due to his current work load he would not be able to contribute meaningfully to the analysis, and suggested that Battelle's computer programs could produce results comparable to those from his.

2.0 1ST REPORT FROM T. BARISH - APRIL 23, 1974

The initial report from Tom Barish was received at the end of April 1974. He had been furnished with all available reports from the meetings held at MSFC. He was also furnished a complete copy of NASA Report 50M23157 which included the analysis and report of the failure of CMG P-11 as well. The first report contained the following analysis and conditions:

1. The #1 CMG failure was a result of overheating and retainer breakup in bearing #1.
2. The cause of this condition was a combination of marginal lubrication, Ball Speed variation, and a possible hangup of the outer race in the slider producing high preload.

An interesting aspect of the analysis (section 10 of report) is the effect on oil migration due to centrifugal forces within the bearing. The analysis suggests the net effect would be a depletion of available lubricant at the inner race - ball and ball-pocket contact areas.

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Battelle had agreed to conduct some testing to evaluate this effect (see section 5.3). The effects of Ball Speed variation (BSV) had been studied as a possible failure mode for P-11 CMG earlier. Under a specific range of radial loads (for a given preload) a maximum in contact angle variation occurs, and in turn produces the maximum BSV and resulting forces in the retainer ball pockets. An analysis can be made to determine the excursions of the individual balls, however there is some question as to the magnitude of the forces generated on the retainer. This force is dependent on the coefficient of friction at the ball race contacts and the value can vary by at least an order of magnitude depending on the lubrication situation. BSV is also considered as a "glitch" generation mode as described in para. 19. A modified ball retainer with increased pocket clearance is recommended. This would allow more ball excursion and reduce the possibility of damage from BSV. It should be noted that this modification will necessitate reduction of the ball complement from 15 to 13. Also recommended are modifications to increase the amount of lubricant available.

3.0 MEETING BGSD, (T. Wheelock) BATTELLE, (J. Kannel) and T. Barish - MAY 7, 1974

The preliminary analysis conducted by Tom Barish and at Battelle had resulted in some areas of confusion and disagreement as to certain aspects of the problem.

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Numerous phone conversations has been necessary; however, it was effectively impossible to arrange a three way discussion. As Tom is unable to travel, it was decided to have a meeting at his office in Van Nuys, Calif. Areas to be resolved were:

- 4.3.1 Critical radial/axial load ratios producing max. BSV.
- 4.3.2 Friction forces producing BSV forces on the retainer.
- 4.3.3 Oil loss from inner and balls due to centrifugal forces.
- 4.3.4 Comparison of failure scenarios for CMG-1 and CMG-2 anomalies.

There was agreement as to the critical radial/axial load ratios for maximum BSV. This range was determined to be 1.5 to 2.0 which corresponds to a slew rate of 3 to 4 degrees per second. The ball retainer forces due to BSV were discussed at length and it was generally agreed the friction forces would be greatly modified by the presence of an EHD lubricating film. BSV however was still to be considered a factor in the failure of CMG #1.

The lubrication problem was also discussed as results of preliminary tests at Battelle to determine effects of centrifugal forces did not indicate this was a

significant factor. It was agreed that testing and analysis of the phenomenon would continue however. Other aspects of the analysis such as race curvature and the relative merits of inner and outer race "control" in this application, thermal consideration such as primary heat generation being at the ball retainer contact, lubrication starvation mechanisms, and design considerations for the "lube nut" reservoir were discussed.

In connection with the redesign of the bearing in order to withstand larger ball excursions in similar application (NERVA) with a design life goal of 40,000 hrs. for a 105 size bearing was cited. The bearing life improved from approximately 500 hrs. to greater than 15,000 hrs. with a similar modification. The modification consisted of removing 1 ball from the assembly and increasing pocket and land clearances. The unit operated at 10,000 RPM.

4.0 SUPPLEMENTARY REPORTS FROM T. BARISH

Shortly after the meeting on May 7, Tom Barish issued two additional reports relating to the discussions and conclusions reached. A graph developed from Battelle's computer program and illustrating the critical radial loads, and resulting ball excursions had been included in their preliminary analysis and in NASA document 50M23157. This analysis in effect supported the BSV failure mode and Tom's report strongly suggests BSV was a prime factor in the failure of P-11,

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Skylab CMG #1 and a likely responsible for glitches in CMG #2. He again recommended the modified retainer design described in his first report as a solution. The third report concerned the effects of inner and outer race ball control on the BSV situation. A modified bearing with reduced outer race curvature had been proposed (principally by Bert Jones) at the time of the P-11 failure analysis. This would in effect maintain all balls in a loaded condition during periods of high radial loads.

This would in turn reduce the tendency for abrupt changes of contact angle during each revolution and subsequent "impacts" between the balls and the retainer. The report suggested that the actual situation is one of partial inner race control and partial outer race control or an oscillation or shift between the two conditions. Included was an approach to determine the actual degree of "control" at the inner and outer races under various loading conditions.

5.0 BATTELLE REPORT - JULY 5, 1974

5.1 Lubricant Migration

This report contains detailed analyses and test results for several important areas in the failure scenarios. Of particular importance is the lubricant migration tests which evaluated the effects of temperature gradients and surface conditions. BGSD had also

conducted investigations into the effects of the orbital environment on lubricant migration (see Appendix 1). The question of what is the fate of oil deposited on the lands of the outer race from the lube nut were of prime importance as this lubricant availability could well determine the life of a marginally lubricated bearing. There is a good chance that a transfer film of phenolic material will occur at the inner race, as the retainer is inner land riding. Any material at the outer race, however, must be wear debris, probably generated in the ball pockets. The possibility of the transfer film combined with the temperature gradient effect in the absence of gravitational forces would indicate lube would not tend to migrate from the lands into the ball grooves.

5.3

Other Lubricant Loss Mechanisms

The results of lube film measurement under various centrifugal loads indicates this is not likely to be a significant factor at the levels which occur in the operation of the CMG. Lubricant loss through the labyrinth seal was found to be negligible. This had also been determined at BGSD during the literature study (ref. MT-15,785). The possibility of lubricant from the retainer being directed onto the lands of the outer race rather than in the ball groove was another important factor in determining the available lubricant quantity. The retainer design and oil distribution are discussed in MT-15,777.

5.4 BSV and Retainer Instability

Reference is made to the computer analysis for critical loading for maximum BSV. It is emphasized that for significant ball pocket forces to occur, a very marginal lubrication situation must exist. The retainer stability analyses indicates no significant instability mode. It is interesting to note the ball impacts occur in situations of marginal lubrication, which may also produce a degree of race control and additional pocket forces. This suggests that the lubrication condition can produce instability as "starved" ball race contacts occur.

5.5 Weightlessness

The effects of lack of gravitational forces were discussed briefly. However some question remains as to the magnitude of the various factors involved. The detrimental effects of lubricant drainage during vertical operation are eliminated as are similar forces which cause flow in the bearing contact areas. Spreading of accumulated oil, dispersion of oil droplets and the behavior of oil in the grooves will also be affected. The operation of the lube nut oil reservoir systems will generally be unaffected, since the operation is dependent primarily on centrifugal forces. It is suggested that further study of the gravitational effect on bearing lubricating mechanisms be conducted.

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6.0 Conclusions

1. The most likely primary cause of the failure of CMG #1 was lubricant starvation.
2. Ball speed variation was probably a factor in the failure of CMG #1 in combination with condition 1.
3. The critical loading for BSV results from an input rate of nominally $3^{\circ}/\text{sec}$.
4. Lubricant deposited on the lands of the races will tend not to migrate into the contact area.
5. Retainer instability is not a primary factor of the failure mechanisms involved.
6. The net effect of the lack of gravitational forces remains speculative and further investigation of this phenomenon is recommended.

APPENDIX I

LUBRICANT SPREADING IN A TEMPERATURE GRADIENT

1.0 FACTORS AFFECTING SPREADING

The distribution of lubricant due to migration in the bearing is affected by a number of factors. They include, among others: (Ref. 1,2,3,4)

1. Temperature
2. Materials
3. Contaminants and Coatings
4. Adsorbed gasses and oxide layers
5. Chemical reactions at the solid, liquid interface
6. Surface finish
7. Porosity
8. Gravity
9. Atmosphere
10. Electrostatic phenomena

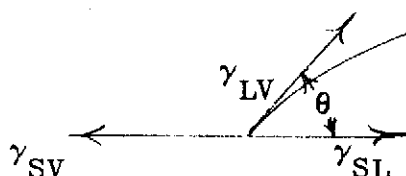
The relative effects of these factors have been studied and some of the more significant analyzed for specific cases but little quantitative data is available.

2.0 ANALYSIS OF SPREADING

The phenomenon of spreading of a lubricant away from a heated area has been observed previously (Ref. 1, P 379). However, no specific explanation was provided for this effect. The spreading process may be studied by the following considerations:

In general a liquid-solid-vapor interface in equilibrium is defined by

$$\gamma_{SV} - \gamma_{SL} = \gamma_{LV} \cos \theta \quad (\text{Young and Dupre' eq.})$$



where

γ_{SV} = Surface energy of solid

γ_{SL} = Surface energy at solid liquid interface

γ_{LV} = Surface tension of liquid

The spreading of this liquid on the solid is defined by a spreading coefficient⁽⁴⁾

$$S = \gamma_{SV} - (\gamma_{SL} + \gamma_{LV})$$

If the surface is wettable S will be > 0 and the liquid will spread.

The effect of temperature on γ_{LV} is described by⁽¹⁾

$$\gamma_{LV} = \gamma_{LV_0} \left(1 - \frac{T}{T_c} \right)^n$$

T = observed Temperature
 T_c = critical temperature

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which shows a decrease in γ_{LV} with increasing temperature (incidentally is approximately = 1.23 for most substances (Ref. 2). This suggests from that spreading should occur more readily in the area of higher temperature unless γ_{SV} increases, and/or γ_{SL} increases at a higher rate than γ_{LV} decreases.

This characteristic change in γ_{SV} and γ_{SL} has not been specifically observed and may be considered unlikely. In the case of a temperature gradient, the surface tension of the liquid may also have a gradient with the lower surface tension at the high temperature edge. This gradient would produce a force tending to draw material from the area of lower surface tension (higher temperature) to that of higher surface tension (lower temperature).

Other factors could also influence spreading in the area of higher temperature. The formation of a preliminary spreading monolayer (ref. 4) would be inhibited due to higher evaporation rates. Evaporation of more volatile components or additions in the area of higher temperature could also produce a lower average surface tension. Tricresyl Phosphate is known to have a high surface tension. If the temperature of the atmosphere is sufficiently below the surface temperature particularly severe gradients may exist in the edge of the lubricant drop. A retraction or negative spreading has been observed in some fluids

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tested on heated surfaces (Ref. 4).

3.0 REDUCTION OF SELF INDUCED TEMPERATURE GRADIENTS

The type of temperature gradient in a bearing is typically one which causes lubricant to flow away from the contact areas. In order to counteract this condition a source of heat remote from the constant region may be introduced. Since the ATM CMG bearing now has heaters, it may be expedient to relocate them in order to generate an opposing axial gradient which would produce flow toward the bearing races.

4.0 REFERENCES

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3. Davies, J. T. and Rideal E.K. "Interfacial Phenomena" Academic Press, New York 1963.
4. "Contact Angle, Wettability, and Adhesion" Advances in Chemistry Series No. 43. American Chemical Society, Washington, D.C. 1964.

THOMAS BARISH
CONSULTING ENGINEER

Bendix Navigation and Control Div.
Teeterboro, N. J. 07608

5411 TYRONE AVENUE
VAN NUYS, CALIFORNIA 91401

(213) 781-8181

April 23, 1974

Attn: Thomas Wheelock, Proj. Engr.

Copy To - Battelle Memorial Institute
505 King Ave.
Columbus, Ohio 43201

Attn: J. W. Kannel

Sky Lab - Bendix Gyros

Bearing Failures

Fourth Report - April 23, 1974

1. References: (a) NASA report number 50M23157; etc. (821 pages)
(b) Three previous reports by TB November 7, 1972, December 31, 1972, and January 22, 1973.

2. Present Problem: To determine as far as possible the specific cause of failure of CMG-no. 1; also the anomalies shown by CMG-no. 2; similar to the ones that occurred before failure of no. 1.

We have available for this analysis the data about the land failure on unit CMG-no. 11, the very extensive and thorough reviewing and testing of this failure. To this we are adding telemeter data from no. 1 and no. 2. This data shows characteristics that must be matched up with the type of failure as follows:

- (a) Torque readings indicated by current which give positive measurement of the size of the additional friction.
- (b) The speed with which the heat builds up. This is important as mechanical action would be rapid, and heat effects would be much slower.
- (c) Periodic variations both high frequency and low frequency.

3. General Conclusions:

- (a) The final catastrophic failure of the bearing including very severe overheating and cage breakup in bearing no. 11 and probably in bearing no. 1, were certainly produced by loss of internal clearance to the point where the thermo expansion of the balls and the inner race being greater than that of the

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3. General Conclusions: (continued)

- outer race developed very large loads. This "runaway" type of failure occurs when the rate of heat generation exceeds the rate of heat dissipation or equalization.
- (b) These bearings operate near the brink of this type of failure because of extremely low oil supply and unusual heat transfer conditions.
- (c) To begin the runaway internal-tightness failure requires some triggering mechanism: candidates -
 reduction of already marginal lubrication,
 sudden large persistent increase in load,
 ball speed variation (BSV) resulting from
 the load combination which produces this, and
 radial bind of the outer race preventing axial
 motion plus over heating of the shaft.

The most likely causes indicated by the evidence are the first and third, and perhaps a combination of them. Each is discussed in turn below.

4. The internal radial clearance of the bearing is too small and not well controlled making the bearing especially sensitive to runaway heat failure. The specifications call for a close control of contact angle but no internal clearance control or control on the race curvatures and tolerances.

Bearing Clearance (Total Radial) (page 8, T.B. report 11/7/72)

	<u>Curvature</u>		<u>Initial Contact Angle</u>	
	Inner	Outer	130	170
Avge.	51.6	53	.00074	.00126
Min.*	51.1	52.5	<u>.00052</u>	.00098
Max.*	52.5	53.5	.00090	<u>.00153</u>

*Tolerance estimated

A thorough test was made heating the inner race to determine how the drag increased when the inner race became hotter than the outer race. The test reported .0017 radial clearance, expected and measured. The test gave the critical temperature difference as over 100 degrees F. With the minimum clearance indicated in the table above, this temperature difference would be about 40 degrees.

- . Furthermore, in operation the balls are likely to be hotter than the inner race and hence would require even less temperature difference. This occurs because the heat transfer does not have any convection, the radiation is small, and the conduction is limited by the small contact area. Therefore, the likelihood of runaway heat failure from loss of clearance is far greater than indicated by the test. This brinkmanship could be avoided by using a minimum radial play of .0015, by controlling the radial play to avoid excessive top play, and by permitting a slightly larger tolerance on the final contact angle.

5. Clearance Measurements on Critical Bearings

(From Records, by phone from Wheelock, 4/24/74)

End Shake	CMG No. 1 (Space Failure)		CMG No. 2 (with anomalies)	
	.0140"		.0144"	
	Brg 1	Brg 2	Brg 1	Brg 2
Make & S/N	Barden 17	Barden 16	SBB 009	SBB 002
<u>Radial Play*</u>	.00091 - .00129	.000969 - .001299	.000873 - .001165	.000719 - .000950
Retainer Looseness on race in ball pocket	.014	.014	.013	.013
	.0055-.0060	.0060-.0065	.0060-.0065	.0060-.0060
Total	.0195-.020	.0200-.0205	.0190-.0195	.0190
Clearance on Brg. O. D.	.000602	.000646	.000425	.000565

* Calculated from measured contact angles and race curvature specifications.

From the above radial play readings, it is quite evident that the two critical units could have had much less radial play than that used in the check test, .0017. The temperature differential necessary to produce a bind would be 40 - 50 degrees instead of over 100 degrees estimated by test.

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- Gyro

- This would be further reduced by the fact the test produced heat inside of the inner race but not at the ball. Note that "delta T" in this case means difference between inner race temperature and outer race temperature and will be used as such in the following. The main report uses this delta T for difference between bearing no. 1 and bearing no. 2 which has been invariably small and not very significant.

6. When delta T begins (temperature differential, inner to outer ring), the first thing that takes place is a reduction in internal clearance, a reduction in contact angle without actual binding. This is equivalent to having a bearing initially made with less contact angle. The table below indicates that the load per ball roughly doubles before binding begins. Hence there should be a gradual increase in friction to about double; an increase in wattage of about 30 to 40 per cent since the bearing friction is not all of it and since you would ordinarily involve one bearing first. The table below shows how a bearing with average tolerances would change. It becomes tight with 78.5° delta T., inner better than outer. With the balls tending to hold the heat, this estimate should drop to about 50° .

Change in Loading with Reduced Clearance

Initial Angle	15°	10°	5°	0°
Angle with 40 lb. Thrust	16.01	12.75	9.9	8.95
Normal Load/ ball	9.62	12.1	15.5	17.1
Initial Looseness	.00098	.00044	.00011	.0
ΔT (est.)	0	43.3°	69.7°	78.5°

contact
angle and
load cancel
out

Further delta T will increase the radial load very rapidly.

Further delta T increase	10°	20°	40°
Added Squeeze (on diam.)	.000125	.00025	.00050
Total added Load (normal)	28 lb.	60	138

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• Telemetry

8. The telemetry from CMG-1 during failure (next page) showed first the difficulty started about one minute after a vigorous manoeuvre. Then the following actions are noted:

- (a) The amperage rose from 1.0 to 1.8 in twelve minutes,
- (b) The amperage leveled off for twelve minutes,
- (c) A short peak, two minutes to 2.4 amps and immediate settling back,
- (d) Leveling off at 2.0 for about one hour

The above indicates that the jump in power is much greater than would obtain with a normal reduction in clearance. The sudden peak might have occurred from the beginning of run-away failure but dropped off too fast for that and is more likely an actual cage breakage. The following high friction would be continued ball-cage or ball-ball friction.

9. Page 7 shows the telemetry for CMG-1 in one of the prefailure anomalies. One very significant condition in this picture and which occurred in the similar anomalies on CMG-2 is the pulsing of the current: maximum + or - .0045 amps. The size and frequency of the change makes it possible that this was a reduction in internal clearance with the contact angle changed as per above. Each peak and drop-off takes two-four minutes and this would be the type of thermal delay that would be evidenced. However, the drop-offs are too sudden and difficult to explain. These pulses could be very small drops of oil entering the ball path. Then the drop-off could be sudden as indicated; however, it is very unlikely to be regular and persistent.

The third possibility is cage bind from BSV which is relieved as soon as heating occurs, changes the contact angle and reduces the increase speed of the unloaded balls. (See discussion later on BSV)

The burst (page 6) to 1.066 amps and the fall off, totaled three hours, is most likely a straight lubrication effect. If there were internal clearance buildup it would show difficulty much more rapidly. This implies a sparse lubrication effect. The bearing temperatures confirm this in that the amperage drops off when the bearing temperature increases which as proved by test increases the oil flow.

The variation in bearing temperature does not help much in determining internal radial bind. In the latter case, it is the delta T (inner to outer ring) which counts. We can only say that an increase in outer race temperature would require a preceding considerable increase of inner race temperature. After this delta T rises, the temperature would equalize in minutes.

Harlow

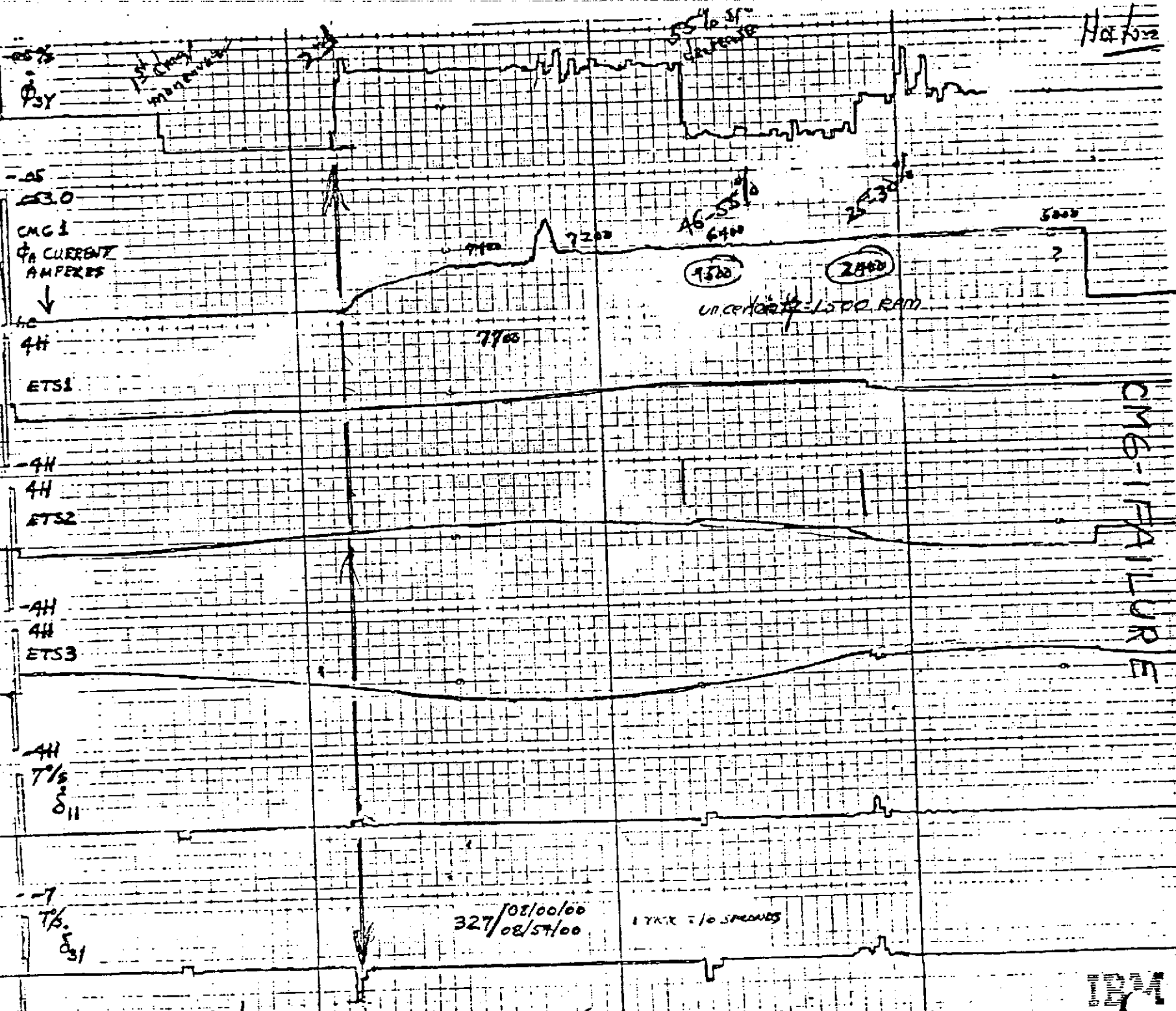
CMG-1 FAILURE

IRM

100%

Total Vehicle Momentum

Pinball
Component



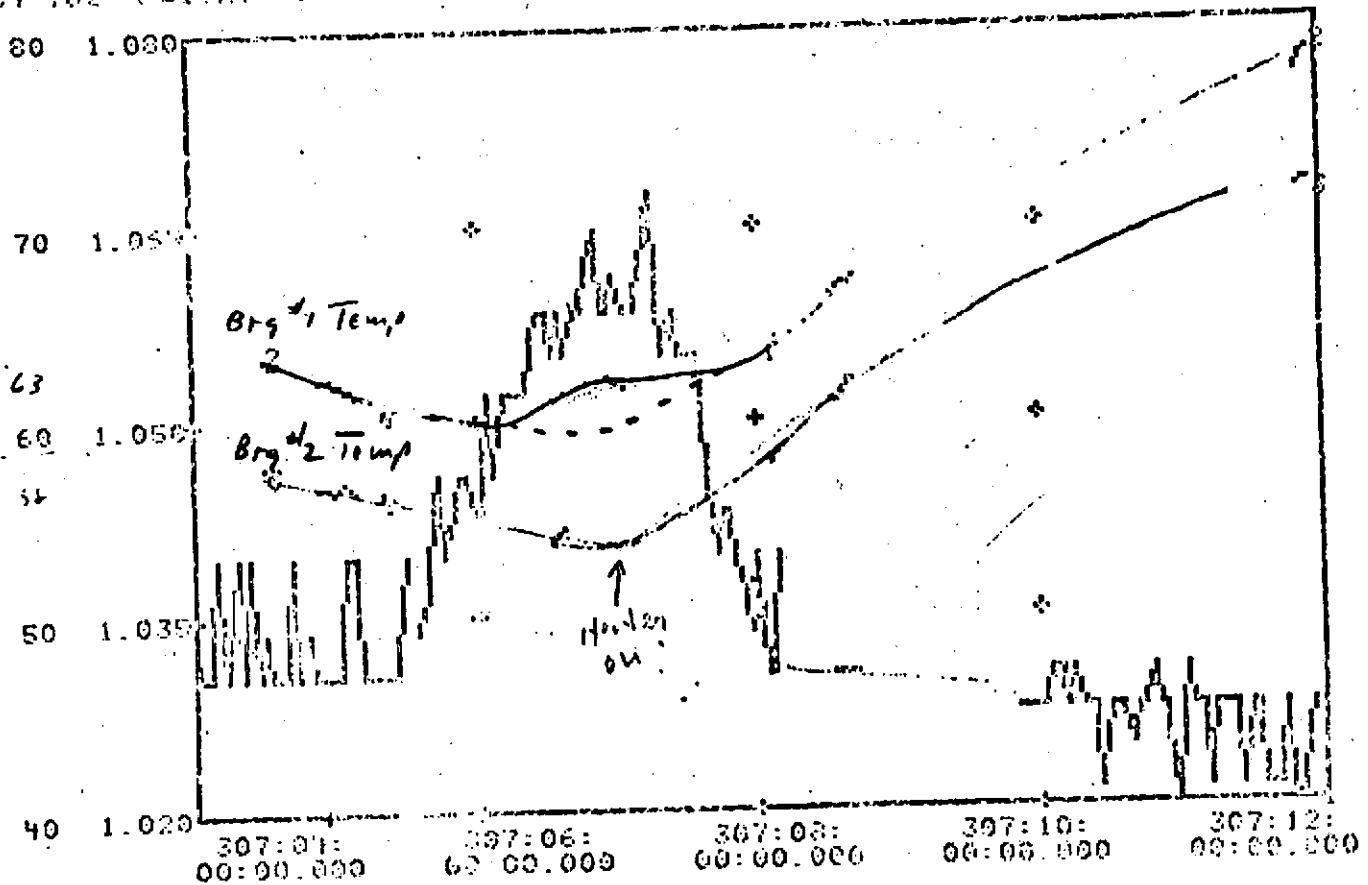
Lewis Cook

CMG - 1 PREFAILURE ANOMALY

REP-1200 NUM-0071 VIEWID-MSF2 SEGS-0002 TO 0002 OF 0003
CMG 1 4A CURRENT VS BEARING 1 & 2 TEMPERATURES

GMT

CMG1 4A CUR CMG1 BRG 1 T CMG1 BRG 2 T
M0404-762 =1(R) 00006-762 =2(L) 00007-762 =3(L)



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#1

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10. Lubrication

The centrifugal force effects need more attention. The inner race is subjected to 2200 "G"s. Hence the oil creep will be far more rapid than under one "G" and directed definitely towards the inner race shoulder. The race can be expected to be bone dry in less than a minute after start of operation. A more exact analysis: the surface tension does not vary with the film thickness. Hence the film thickness can be determined as that which would just equal the surface tension at 2200 "G"s. However, there are too many questions involved. It would be better to make a simple test and find out how much film remains and how long it remains under the high CF. Note that this oil would be thrown off the edge of the inner race and would tend to provide a certain minimal lubrication for the bore of the cage.

11. The oil delivered by the nut should definitely be much greater at the first of the run because the pressure developed by CF on the metering device varies with the depth of the oil in the nut. However, this need not modify previous conclusions since we actually tested the oil flow at different depths.

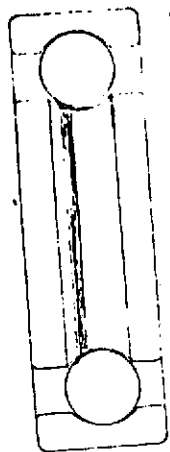
12. The oil leaving the nut and entering the cage lip has a CF of 390 "G"s hence the cage is practically bone dry in the pockets except for whatever the ball transfers from the outer race to the cage and the inner race. There is also possibly slight lubrication from the inner race throwing into the cage but even this will mostly go through the cage pockets. Hence the unusual squealing under severe loading because of the dry contact. However, producing a squeal requires positive forces on the contact and these are most likely from ESV.

13. The high CF on both the nut and the cage means that oil flow from them will take place strictly in a tangential direction at high speed and at the very first available opening. The location of the nut lip with respect to the cage was determined as .006 minimum overlap. However, this figure did not include an allowance for the axial motion of the cage (.004 off-center possible) and the fact that the ball center is not in the same position axially as the groove of the outer race (.002). This indicates that with some tolerance variation and wear the oil from the nut could skip the cage entirely. The oil leaving the cage oil holes would move outward radially at the first small edge and this would clearly hit the outer race bore outside of the ball groove. It would take an appreciable creep in the outer race under zero "G" to reach the ball path and some of the oil would definitely creep in the opposite direction.

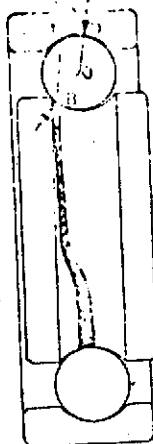
14. • The sparsity of oil on the inner race (practically none) will make the friction at that contact considerably higher than normal and definitely higher than the outer race. The A.B. Jones computer setup uses .07 friction as carefully determined by tested calculations for normal oil and normal bearings. This inner race will develop more friction and more heat. The outer race can have an elastohydrodynamic (EHD) film. There will be two problems: 1. the removal of oil by the balls and feed to the inner race and the cage. Also creep of oil away from the contact will starve it unless the creep brings the oil back relatively rapidly. Both of these effects will leave so little oil in the race that its film thickness may be less than required for an EHD film. A rough calculation assuming the average oil flow and one minute for removing the oil by CF and creep leaves a film thickness of .0004. This should be ample for an EHD film but the calculations are very rough. Furthermore, the radial load and the load per ball becomes 5-10 times as large under violent manoeuvres. We should check the EHD film capabilities at these conditions. This is one likely candidate for the pulsing of the friction.
15. The enormous amount of thorough testing of the metering system, the amount of flow and what it does to the bearing leaves little to add.
- We add one note: the difference in oil flow that exists from the maximum to the minimum temperature allowing at the bearing outer race, can be quite critical in view of the fact that the remaining oil film is extremely small. It is quite possible that at the low rates, the inner race very soon becomes bone dry with much higher friction and much heat added to it.

Ball Speed Variation (BSV)

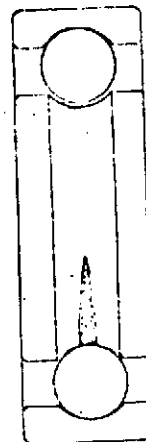
(Ref.: "BSV In Ball Bearings and Its Effect on Cage Design" ASLE Paper 68 AM6C-2)



Thrust load only.
No appreciable BSV



Thrust and radial
Approximately equal
Major BSV occurs

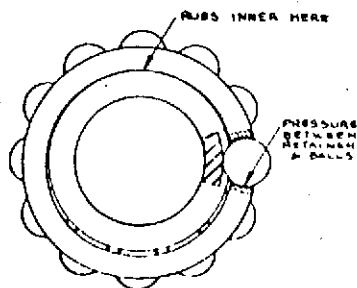
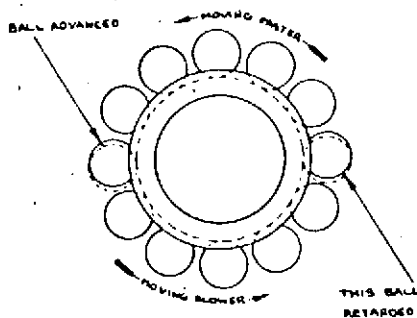


Radial load mainly.
No BSV binding.
Upper balls loose
(except for CF)

A.B. Jones Computer setup for earth conditions gave a contact angle variation (center figure above) of 27 degrees. (T.B. 3rd report, Jan. 22, '73)

It is urged that computer runs be made for space conditions. This should be 40 lb. thrust and radial varying from 20-60 lb. in small steps. We need to define the BSV which exists only for a narrow band and also how large it will be. Indications are that it will be maximum at about 40 lb. radial load; $1/2^\circ$ per second gimble motion.

16. BSV Forces:



The BSV makes the cage operate eccentricly (see figure). For small BSV the cage floats in its most comfortable position and the increase in friction is practically nothing. However, when the ball excursion exceeds the total looseness of the cage (cage-race and cage pockets), the forces between the cage and balls and race rise very sharply. With our flexible cage there is a small band when the cage yields and this determines the size of the forces but they soon reach a magnitude of 5-20 times the normal bearing friction. With metal cages, this is sudden.

17. In these gyro bearings, the total available looseness is smaller than usual:
.0190 - .0205
for the critical space bearings. (see page 3)
The computations for earth conditions showed a maximum ball excursion .032", for outer race control. This would give positive binding. We do not yet know what the computation will give for the space conditions.
The BSV is sharply different in the critical areas when the outer race controls the ball spin than when the inner race controls the ball spin. The computer setup gives the ball excursion for the early conditions, but in this critical area where it is not clear which race controls, the computer does not give the ball excursion but it does give the contact angle. T.B. report no. 2 and 3 computed the ball excursion for both conditions.
Furthermore, the computer assumes .07 constant coefficient of friction for both races. The inner race, when it becomes bone dry due to CF will surely have a higher coefficient of friction.
18. Cage Instability would not normally be noticed with the very light and flexible bakelite compound cages unless there was a very definite bouncing with considerable force behind it. It is the author's belief (not confirmed by tests) that this occurs during the unstable condition when the bearing may be rapidly changing from inner-race control to outer-race control. This would give a considerable forced vibration that could be perceived, and would explain this squealing.
19. If the BSV produced cage bind, the current pulsing of CMG-2 would have occurred as follows: the onset of rapid heating would first reduce the internal clearance. This would happen even more rapidly than from normal load because the heat is generated at cage-race contact and cage-ball contact. The balls would heat up more and more rapidly. The reduction in clearance would quickly reduce the effective contact angle (table, page 4). This would sharply cut the BSV since the high angle at the top of the bearing (middle figure, last page) would come down rapidly. The elimination of BSV bind would then permit an equalization of the temperature and a drop in friction. This would also explain the "glitches" as the speed would rapidly drop and come back more slowly.
20. Outer Race Bind
The housing around the outer race of the bearing has .12 thickness of beryllium bronze and .83" of aluminum. Any changes with temperature would closely follow that of the aluminum. The clearance here is .0005 nominal. Actual clearances in the critical bearings was .00043 - .00065 (page 3).
If the unit were measured and assembled at 82°, operation at 60° would completely remove the minimum looseness above and make the unit bind. This would happen much more quickly if the aluminum were colder than the outer race.
Such binding would quickly prevent any expansion axially of the shaft.

- and build up large thrust loads: 50° delta T gives .003" expansion and would increase the thrust load from 40 lb. to 700 lb.
This difficulty is unlikely: first, because both bearings would have to bind. Secondly, the aluminum housing would expand also so we would need a differential. Nevertheless, steps should be taken to prevent its possibility on future designs. Suggestions: slightly increased radial play perhaps .0008. Also the measurement of the aluminum housing should be specified at a particular temperature.

21. Temperature Differences on Outside of Gyro Housing.

If one side of the gyro housing is in sunlight (not the end-bells) and the other side in the shade, the difference in axial expansion would produce a tilt of one end-bell with respect to the other, and a misalignment on the two ball bearings. As explained above, these bearings are especially sensitive to misalignment because of the rapid increase in BSV. This problem is very remote because the gimble housing is shielded. Even a 100° difference in temperature would give only .0004"/" misalignment on each bearing.

Summary

22. This entire analysis is only of value for assuring better performance on future gyros. The following actions have been suggested:
- (a) Compute the BSV for space conditions.
 - (b) Increase the bearing looseness a small amount but control the looseness rather than the contact angle. Also specify a control on the race curvature. A flatter race curvature on the inner race particularly would permit an increase in looseness without much change in the contact angle.
 - (c) The cage needs many adjustments:
The cage pocket should be elongated circumferentially about .05. This will require cutting from 15 to 13 balls.
The cage-race clearance can be cut to the original .005 - .009. Increase width to better its oil catcher.
The oil holes in the cage should come nearer to the center.
The scallops on the bore should leave an appreciable flat between each of them since the pointed condition now gives poor bearing and easy removal of cotton slivers.
 - (d) The OD looseness should be slightly increased.
 - (e) Changing the curvatures should be checked carefully. Any attempt to flatten the race curvature on the inner race and leave the outer race controlling the ball spin, would also mean more heat generated at the inner race and greater delta T. Nevertheless,

Summary (continued)

it may be desirable.

- (f) The oil flow should be checked to avoid very low figures as occurred in some of the space bearings. Also perhaps, closer temperature control.
- (g) The nut OD can be increased about $1/8"$. This would permit about $1/16"$ larger oil space and would require moving the metering out also.
- (h) Retest to confirm no additional trouble from all the above changes.
- (i) It may be desirable to again attempt to reproduce the symptoms obtained in space by running a unit with all the unfavorable condition discussed.

Thomas Barish

Ball Speed Variation in Ball Bearings and its Effect on Cage Design

THOMAS BARISH, ASLE
Consulting Engineer
Van Nuys, California



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Ball Speed Variation in Ball Bearings and its Effect on Cage Design

THOMAS BARISH, ASLE
Consulting Engineer
Van Nuys, California

A review is presented of the common causes of Ball Speed Variation (BSV), with emphasis on misalignment, the most common cause, and including typical calculations.

Three very different stress regimes result, depending upon the extent of BSV. When large, rapid cage failure is produced by the large forces at the cage-ball and cage-race contacts.

In many cases, the trouble can be cured by small changes in the cage design, particularly increases in the looseness at the ball pocket and the race. Typical field experience is quoted.

INTRODUCTION

Ball Speed Variation (BSV) refers to the speed of the ball along the pitch circle. A. B. Jones aptly names it "orbital velocity" and gives a complete set of equations (1) (2). Variation occurs in practically every operating ball bearing, but fortunately, rarely large enough to be serious. When large, it produces rapid failure at the cage-ball and cage-race contacts.

BSV has been known for a long time; i.e., Fig. 1, a production cage design used by D.W.F. (Germany) about 1920; clearly the designer expects changes in the ball-to-ball distance. The author's first contact with this problem was in 1924: a 420 size Conrad type bearing under heavy thrust and radial load at moderate speeds showed excessive temperature rise and roughness. A bearing without a cage was put in a testing machine

under the same loads and turned slowly by hand. The distance between the balls was measured with an inside micrometer, and varied up to $\frac{1}{16}$ in. Removing the cross-sectioned area A in Fig. 2, without any other change in the bearing, eliminated the roughness and reduced the temperature rise about 75F.

Cause and Types of Stress

The orbital velocity of the ball is one-half the velocity of the contact point B on the inner race, Fig. 3 (one exception at high speeds with large centrifugal force is described later). The only factor in the equation which can vary in operation is the contact angle. Hence all BSV results from contact angle changes.

The cage rpm is controlled by the balls and equals the average ball speed. The balls advance or retard from the average speed and crowd the cage pockets and the cage. This is the major source of severe loading at cage-ball and cage-race contacts: possible other source, severe unbalance.

These stresses are not controlled by the BSV itself, but by the change in the ball position, the displacement variation. Most variations in speed are cyclical: the ball will travel faster than average in one-half the bearing and slower in the other half. The advance of the ball accumulates over half a revolution. To determine the ball position and the distance it departs from the average requires an equation or curve for the ball speed variation, and an integration for the ball position variation from average.

The ball-cage forces can have three radically different

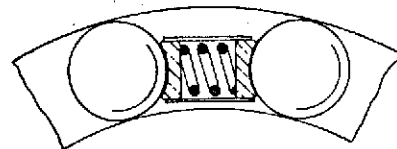


Fig. 1—Spring cage—circa 1920 (D.W.F.—Germany).

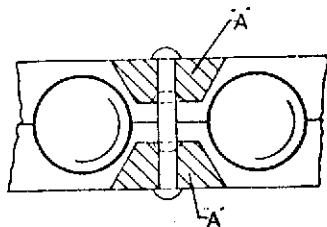


Fig. 2—Removing metal A dropped temperature about 75F (1924).



Fig. 4—Cage wear from BSV.

regimes: (a) When the ball displacement is less than the looseness in the cage pocket and no cage-ball bind results. Only two small forces remain: the relatively negligible cage weight, and the unbalance force appreciable only at the highest speeds. For the 7204 bearing the cage weight is only 0.013 lb for a suitable lightened machined cage (0.009 lb for a stamped cage). The unbalance force at 75000 shaft rpm (1.5×10^6 DN) will be 4.7 lb for 0.015 eccentricity (0.030 loose), and drops to 0.60 lb at 27000 shaft rpm. These unbalance forces are small and occur only when the cage eccentricity rotates.

If the balls are not initially spaced in the middle of their travel, considerable forces arise to push them into median position, but these exist only for the first few revolutions. The cage should be sturdy enough to avoid failure under these momentary forces. (b) *Moderate Cage-Ball Forces* arise when the ball displacement exceeds the clearances. The cage must yield by an amount equal to the difference between displacement and clearance. The force developed varies with this difference and the spring rate of the cage circumferentially.

The cage revision of Fig. 2 sharply reduced the spring rate and the cage-ball forces. Most cages have high spring rates circumferentially and large cage-ball forces in this regime of moderate BSV. Hence rapid failures occur especially above moderate speeds. Stamped cages are better than machined cages, and there is room for much ingenuity in making them soft circumferentially but sturdy as a ring. A spring cage like Fig. 1 or those used in gyro gimbal bearings keeps these forces low, but such cages are usable only for low speeds.

The usual cage materials have low fatigue limits and fail soon under these conditions. Hence the range of BSV that can be handled by flexing is quite narrow.

Cage materials, especially the bronzes, are selected so they can spread or coin or wear at the cage-ball contact without immediate failure under average conditions and without transferring metal from the cage to the balls or races. For small BSV, the cage pocket wear and increased looseness can soon exceed the BSV and the large forces disappear. The same action takes place at the cage-race contact but to a much smaller extent, i.e., Fig. 4. (c) When ball displacements are large and cages stiff, the balls are forced to skid or slide on the races. Cage-ball forces are 10 to 100 times larger and failure very rapid. The force size is indicated by Palmgren data (3) of Fig. 5, where a ball was forced at right angles to the direction of rolling. With BSV, the ball is forced in the direction of rolling, either faster or slower than the rolling speed; the forces will be nearly the same. Above 0.4% sliding-to-rolling, the friction is large enough to induce gross sliding. Indicated coefficient of friction can be up to 0.25, probably sliding on one race first. With the 2 to 1 leverage, this gives ball-cage force up to 50% of the ball load.

At very small amounts of skid, actual sliding does not occur. Instead, the ball and race materials yield in

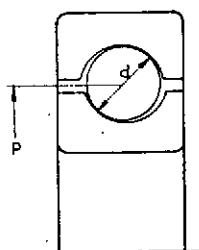


Fig. 3—Ball speed formula: $n = \frac{1}{2} N [1 - (d/P) \cos a]$

where n = orbital velocity, rpm
 N = shaft rpm
 d = ball diam. in.
 P = pitch diam. in.
 a = contact angle

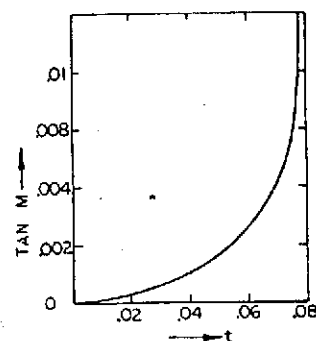


Fig. 5—Ball rolling between flat plates with side force.

M = angle ball travels from pure rolling,
 = ratio cross motion to rolling
 t = Tangential stress/perpendicular stress

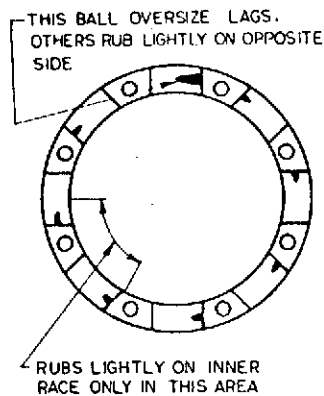


Fig. 6—Cage wear from oversize ball.

shear parallel to the surface and below it. This absorbs much less power than sliding. For a more complete discussion of this effect, called "surface shear", see reference (4).

One Large Ball

Most BSV effects are cyclical and repeat exactly each revolution. One rare exception occurs when one or more balls are appreciably different in size from the rest of the balls in the bearing. The oversize ball will have a smaller contact angle and lower speed, and will lag continuously. In a 7204 bearing, a 0.0001 in. oversize ball reduces the contact angle from 35° to 33.5° , and causes the ball to lag 0.0022 in. per cage revolution. At 3600 rpm, this would accumulate to 3.2 in per minute. Hence the greater importance of ball size equality in angular contact type ball bearings.

Failures produced by this condition have one pocket with severe ball-cage stress on one side of the pocket (Fig. 6), and all other pockets will have lesser stress on the opposite side of the pocket; also, mild rubbing on the inner race, localized on the cage, not on the race.

THRUST LOAD PLUS MISALIGNMENT

Misalignment is the most common cause of BSV and cage failures. Under pure radial load no unusual cage-ball distress occurs because the balls readjust their position in the unloaded area. BSV problems arise only with pure thrust load, or enough thrust to keep all the balls in contact (also in very high speed bearings, discussed later).

Figure 7 shows ball path pattern for thrust load plus misalignment in an angular contact bearing with inner race off-square: Fig. 8 with outer off-square. Figure 9 shows an inner race after operation. No real failure occurs in the race contact. The indicated contact angle change does not cause failures in the races or the balls, no more than they would operating in a bearing designed to such angles. The only difference at the contact is a small reduction in fatigue life, because the load distribution among the balls is modified. The failures are primarily in the cage.

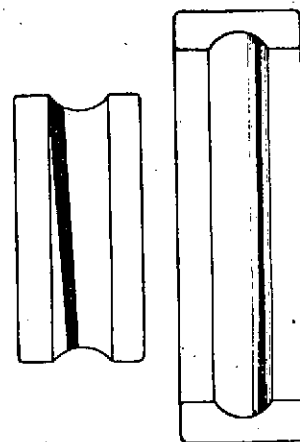


Fig. 7—Contact patterns—with off-square inner.

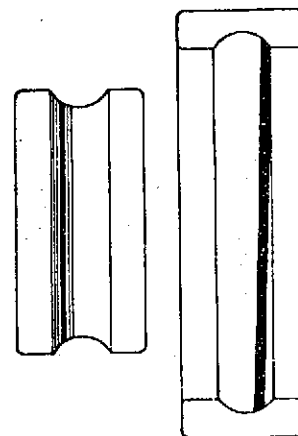


Fig. 8—Contact patterns—with off-square outer.

The balls travel faster in the upper half of the bearing, and slower in the lower half, (Fig. 10), and the ball position displacement accumulates as shown. This forces the cage downward to an extent equal to the maximum amount of ball displacement from average. The displacement of the ball from the average position is approximately sinusoidal. If the cage is allowed to operate in an eccentric position as shown in Fig. 10, the ball pockets also provide a sinusoidal displacement from average. Since the cage is not fixed except by its clearance from the race, it will adjust itself to this position and no unusual cage-ball stresses occur, only the weight of the cage. If the eccentricity rotates with the inner race, there will also be an unbalance.



Fig. 9—Off-square inner ball path tilted—no real race failure, cage broken.

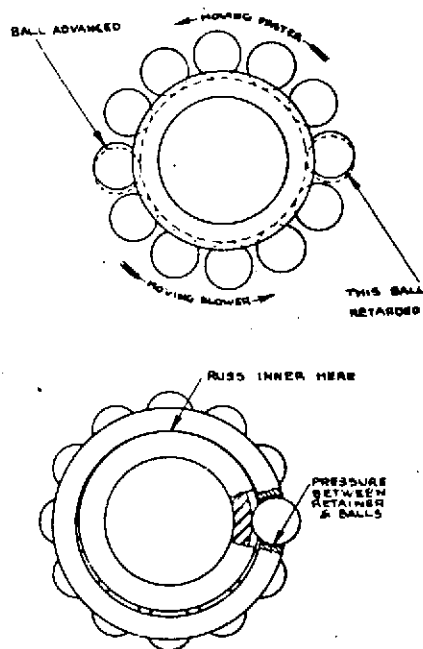


Fig. 10—BSV moves cage off-center.

Table 1 gives typical figures for ball position variation under misalignment.

It is evident that the misalignment capacity of such a bearing can be materially increased by increasing the looseness of the cage, both in the pockets and on the inner race. If the total looseness in both locations were increased from the normal of about 0.015 in. to 0.04 in., the aligning capacity would be increased from 0.11° to 0.29°. The cage-pocket and cage-race loads would then be in regime (a) above, just cage weight and unbalance for inner off-square: no heavy loads as in regime (b) or (c).

The total clearance can be divided between the ball pocket and the cage-race contact, except that a minimum is needed in the pocket for departure from true

TABLE 1—BALL POSITION VARIATION vs MISALIGNMENT

7204 Bearing: 10-1½ balls, 35° angle 52 and 52% curvature				
Misalignment in./in.	0.001	0.002	0.003	0.005
degrees	0.057	0.115	0.172	0.29
Contact Angle Change from Average, degrees	±3.44	±6.82	±10.2	±17
Ball Position Variation from Average, in.	±0.0038	±0.0076	±0.011	±0.020

Note: Position variation will change up to 50% with size of load, or with a small radial load; and the latter depends on relative direction of misalignment and load. Displacements shown are only approximate; no computer setup used. Also BSV assumed sinusoidal.

sinusoidal motion. Large pocket clearances would give cage instability axially. Hence most of the increased clearance should be at the cage-race contact. Also satisfactory would be cage pockets elongated circumferentially, but these are difficult to make except in a two-piece riveted cage where temporary spacers between the halves will give elongated pockets.

Stamped ribbon-type cages, riding the balls, automatically leave room for liberal clearance on both the inner and outer race. However, they are frequently made with the inner and outer clearances unequal limiting the alignment capacity to that provided by the smaller opening.

The same type of BSV but of lesser magnitude results from: (a) excessive angular deflection of shaft or housing, (b) moderate radial loads with larger thrust loads, large enough to keep all of the balls in contact, and (c) off-center loads or moment loads on a single bearing mounting. In all of the above cases, the ball displacements rapidly increase with closer curvatures and with higher contact angles.

HIGH SPEED WITH COMBINED LOAD

BSV becomes serious at high speeds when the centrifugal force of the balls is appreciable (usually above 500,000 DN), and when radial loads are combined with small thrust. With pure radial load, the upper balls are held against outer race (Fig. 11a) and the cage force is only that needed to drive these balls. The looseness C increases to initial looseness + deflection of bottom ball + deflection of top ball at outer contact. The first small added thrust load makes little change until the axial motion brings the top ball into contact with the inner race (Fig. 11b). Because of greater looseness this occurs at a high contact angle and produces the peculiar ball path shown. The top ball rides the outer with a low angle, 5° to 10°, and the inner angle can easily reach 30° to 40°. The higher angle increases the ball speed.

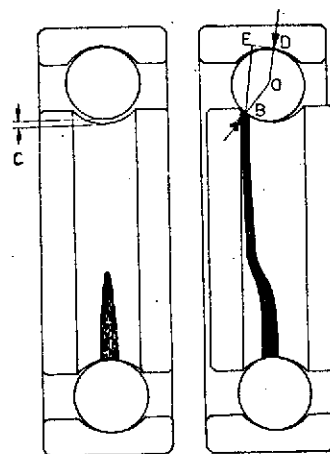


Fig. 11—Ball path patterns at high speeds—(a) Pure-radial load; C = Initial looseness + Deflection at 3 contacts. (b) Adding thrust moves outer to left. Top ball touches inner at B with high angle. Ball speed = (Speed of B) (OD/BE).

It is now more than half the speed of the driving contact (Fig. 11b). Different angles occur also with pure thrust at high speeds but they do not change: hence no BSV. Formulas and experiments are reported by Hirano (6). The BSV becomes large under combined load because the lower balls still have a small contact angle. A. B. Jones' equations (2) and computer programming give the change in angles and orbital velocity.

Details are given for an early case of trouble from this source (1954), the turbine bearings on an aircraft alternator drive. No trouble arose in extensive testing under load with 80 lbs thrust, but there were repeated failures in qualification test at no load with only 20 lbs spring thrust. The higher operating thrust moved the outer axially and reduced the BSV by reducing the inner contact angle at the top ball. The problem was solved quickly by increasing the spring load from 20 lb

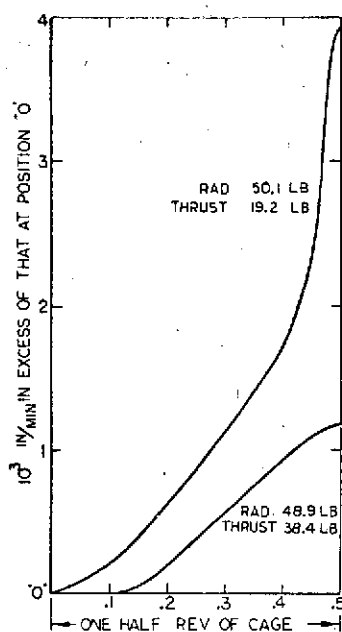


Fig. 12—BSV for conditions of Table 2.

TABLE 2—BSV IN TURBO-ALTERNATOR DRIVE AT NO-LOAD

Bearing 204 Conrad—8-5/16" Balls 51.6 and 53% Curvatures—0.0006" Loose Cage Looseness: Pockets 0.008—Outer Race 0.010 Conditions: 25,000 rpm—Radial Load 50 lb Centrifugal Force—6.5 lb per ball				
SPRING THRUST	19.2 LB		38.4 LB	
	MAX.	MIN.	MAX.	MIN.
Contact Angles				
degrees—Inner	36.3	13.7	24.4	16.5
Outer	0	13.3	9.7	13.7
Ball Position				
Variation, in.	±0.019		±0.0088	
Total	0.038		0.0176	

to 40 lb. The calculated BSV for both loads are given in Fig. 12. The total ball position variation became less than the total cage clearances (Table 2). Increasing the cage clearances would also have eliminated the trouble (as was done in later cases), but would have taken more time.

Table 3 gives data for a later similar case (1964) with angles and orbital velocities by computer, and corrected by increasing clearances.

TABLE 3—BSV IN TURBO-PUMP

Bearing 305 Conrad—9-1/2" balls, 25,000 RPM
17° Initial angle: 51.6 and 53% curvatures
Loads: 359 lb radial; 175 lb thrust; 32 lb C.F./ball

Figure 13 shows BSV and position variation from average, and calls for ±0.055 cage looseness, total for pocket and race contacts. Only part of this can be compensated by eccentric operation of the cage; that part

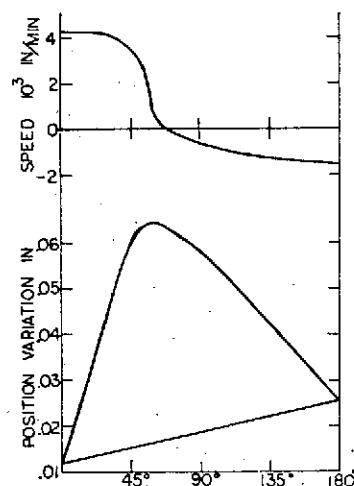


Fig. 13—BSV for bearing of Table 3. Speed is variation from average. Position from arbitrary starting point.

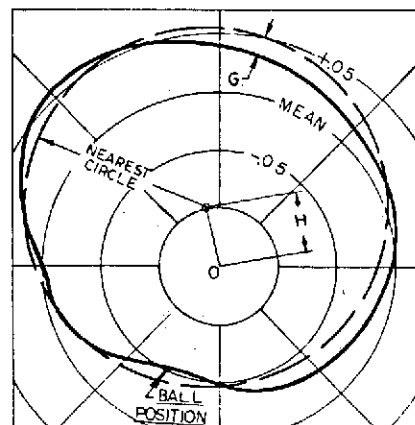


Fig. 14—Determination of cage clearances. G is minimum needed at ball pocket. H is maximum usable at race.

which gives a true sinusoidal ball position variation. The remainder requires pocket clearance. To find the nearest sine curve for Fig. 13 is time consuming and inaccurate. It is better to use polar coordinates Fig. 14, plotting the circumferential ball position variation as the radial coordinate. The nearest circle shown gives:

- (a) minimum acceptable pocket clearance as variation from curve to circle, $G = \pm 0.015$
- (b) maximum useful cage-race clearance, as off-center of circle, $H = \pm 0.040$

Since race clearance might cause unbalance troubles, the pocket clearance was set above the minimum ± 0.015 and the cage-race clearance set for the balance of ± 0.055 total.

Minor misalignment or angular shaft deflections will change BSV under above conditions rapidly. They can add or subtract according to angular relation between the radial load and the misalignment.

Bearings that have operated under these high BSV do not show the ball path pattern of Fig. 11b. The rotation changes this pattern to an extremely wide path with fuzzy edges. The pattern was evident in only one case where the radial load rotated with the inner race.

CONCLUSIONS

Appreciable BSV arise in ball bearings, notably under (a) misalignment plus pure thrust load or predominant thrust load, and (b) high speeds with radial load and small thrust. Examples show the extent of BSV and ball displacements; also methods of determining them, and corrective measures used. Increased cage clearances can permit much larger ball displacements from these sources without generating high cage forces and early failure.

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- (6) Hirano, Fujio, "Motion of A Ball in Angular-Contact Ball Bearings" *ASLE Trans.* 8, 1965, p. 425-434.

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THOMAS BARISH

CONSULTING ENGINEER

Bendix Navigation and Control Div.
Teeterboro, N.J. 07608

5411 TYRONE AVENUE
VAN NUYS, CALIFORNIA 91401
(213) 781-8181

Attn: Thomas Wheelock, Proj. Engr.

May 9, 1974

Copy To - Battelle Memorial Institute
505 King Ave.
Columbus, Ohio 43201

Attn: J. W. Kannel

Subject: Sky Lab Gyro Bearings

1. Reference: The useful and satisfying conference at this office with both of the above men on May 7th.
2. The curves developed at Battelle for the ball excursion under space conditions (next page) are exactly what the author requested in T.B. 4th report, Apr. 23, 1974 (also suggested in 1st report, Nov. 7, 1972, 2nd report Dec. 31, 1972 and 3rd report, Jan. 22, 1973). I regret having overlooked the considerable importance of this set of curves in the first review of the long report.
These curves give the following conclusions:
 - (a) A considerable BSV (ball speed variation) occurred under space conditions:
 - (b) The resulting ball excursion was approximately twice the total looseness available in the cage for this movement.
 - (c) Relatively large forces were generated, enough to produce the "squeal" and the "glitsch".
 - (d) The heat generated would be enough to rapidly reduce the clearance, and the high contact angle for the ball opposite the radial load (ball no. 1). The reduced angle would immediately remove the BSV bind, and the expected period for this heat action would be of the same order of magnitude observed in the "anomalies": about 4 min. cycles:
 - (e) The continued heat generation would trigger a complete thermal bind failure.
3. It is still desirable to investigate what happens to the size of the BSV with outer-race-control instead of inner-race-control.
Just before the radial load becomes large enough to leave ball no. 1 completely unloaded, the upper balls will certainly be outer-race-control because the load on the inner-race will be very small whereas the outer race load is still the full centrifugal force. This means the lower part of the bearing will be inner-race-control contact and the upper part the opposite.

BSN
Contact Pressure 10^3 psi

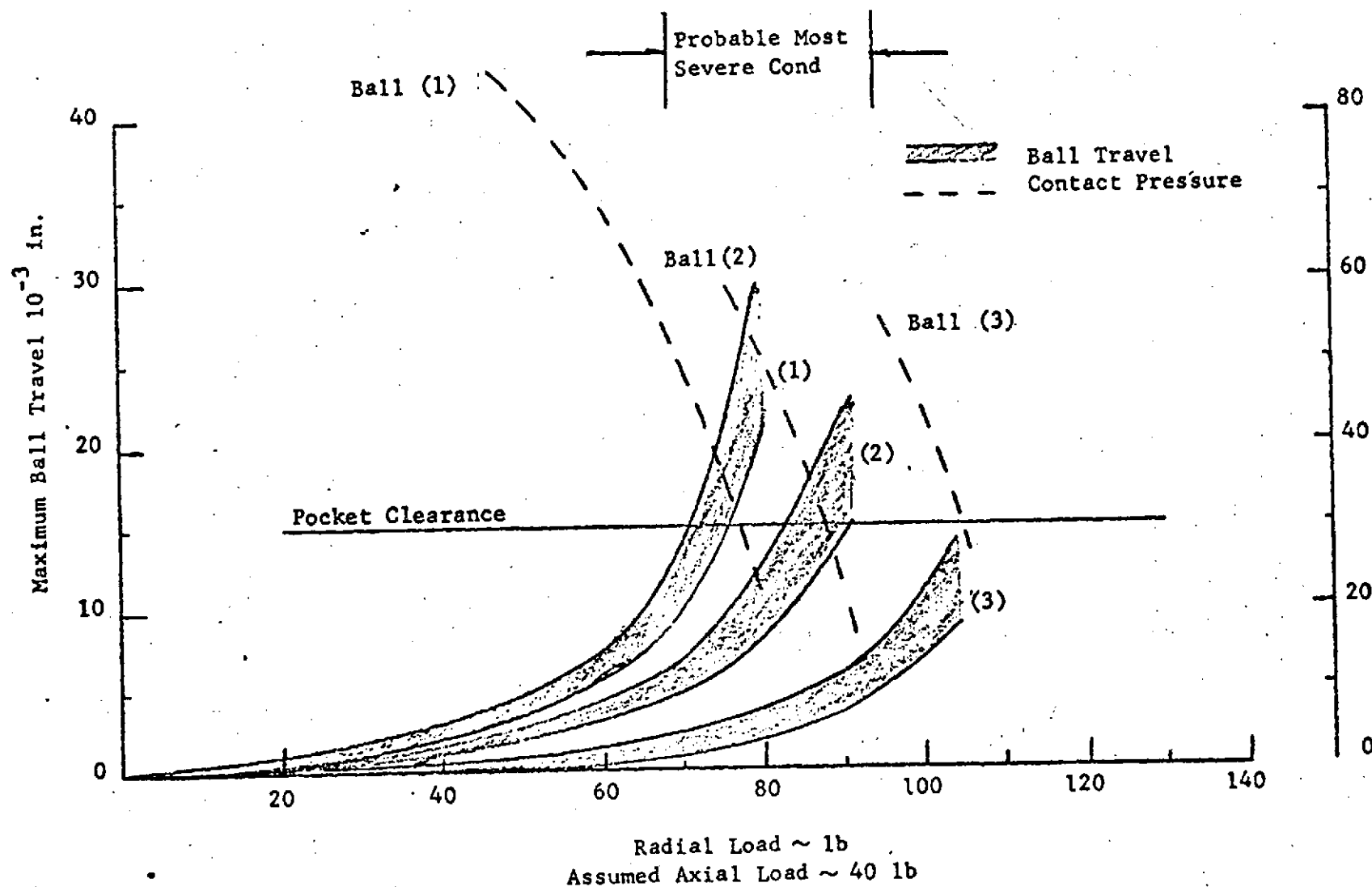


FIGURE 2. SUMMARY OF COMPUTER PREDICTIONS OF BALL TRAVEL IN CMG BEARING

4. We can now draw the positive conclusions that the initial cause of the failure in CMG-11 and CMG-1 and the anomalies in CMG-2 was the ball speed variation. The final complete failure was induced by the subsequent elimination of internal clearance and the considerable thermal bind: though it is still possible that the BSV actually broke the cage and this induced the final failure. It would have been helpful if we had run this additional check over a year ago.
5. The cage stability papers by Kannel and Walters are a truly admirable and a thorough treatment of the cage motions: a virtuoso computer performance. The reports do not indicate it, but I presume it includes a full calculation of BSV. For our work there are only two of these motions and forces that are appreciable, the cage-race bind and the cage-ball bind. It is enough to know that these forces are large enough to produce the "squeal", the speed-drop, the heating, and ultimately the binding failure. The BSV bind forces would be changed by variation from inner-race-control to outer-race-control and by probable variations in the coefficient of friction. However, this does not change the conclusions appreciably as long as there is a failure. The forces are largely limited by what it takes to make the ball slide on the race.

Fortunately the problem can be completely solved by the suggested redesign of the retainer: circumferentially elongated pockets with reduced clearance at the cage-race contact and axially in the pockets; so that the stability motions are sharply reduced. This will eliminate all of the BSV binds, except for the one revolution of the cage (15 milliseconds) it takes to slide the ball into the available free space.

6. A further reduction in the already sparse lubrication could have been an independent source of failure. The timing makes this very unlikely: the start in CMG-1 at violent motion and the 4 min. cycling. However, the lubrication problem may have been definitely contributory as did happen with the reduced radial clearance.

Thomas Barish

SUMMARY REPORT

on

TECHNICAL SUPPORT TO NCD OF
BENDIX CORPORATION ON ANALYSIS OF
SKYLAB CMG ANOMOLIES

to

BENDIX CORPORATION

July 5, 1974

by

J. W. Kannel, D. K. Snediker
J. W. Kissel, and S. S. Bupara

Contract No. PO 88-451350

BATTELLE
Columbus Laboratories
505 King Avenue
Columbus, Ohio 43201

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TECHNICAL SUPPORT TO NCD OF
BENDIX CORPORATION OF ANALYSIS OF
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SUMMARY AND CONCLUSIONS

The project was conducted in four tasks directed at clarifying the failure of CMG No. 1 and the abnormalities seen in CMG No. 2 of the Skylab spacecraft. Task I involved experimental evaluations of lubricant migration due to creep against a thermal gradient and in the presence of wear debris and polymeric transfer films. Task II involved evaluations of lubricant leakage through the labyrinth seal. Task III was concerned with elimination of discrepancies between ball speed variations (BSV) predicted at BCL and those predicted by a consultant, Mr. Tom Barish. Finally, Task IV involved an evaluation of the propensity of the retainer to become unstable.

In general, the research substantiated the earlier conclusion that the primary cause of failure was a result of lubricant starvation. Other effects such as BSV could well have contributed to the demise of the starved bearing. The specific conclusions from the four tasks are as follows.

Task I. Lubricant Migration Evaluations

The prime lubricant in the CMG will not creep efficiently across surfaces prerubbed with impregnated synthane. Synthane wear debris also acts as a creep barrier. This suggests that the ball track area cannot be replenished by creep.

Analyses indicate a temperature difference of 6 F exists between the ball-track area and the shoulder of the CMG bearing. Our tests show that KG-80 will not creep from cold to hot against as little as a 4 F gradient. Thus, a relatively hot ball track will cause oil to migrate away and lubricant replenishment by creep will be retarded.

Task II. Labyrinth Seal Investigation

The estimates made of lubricant flow rate through the labyrinth seals indicate that the loss by this means is negligible with respect to the lubricant supply rate to the bearing. Therefore, it is not a factor in lubricant depletion from the bearing.

Task III. Investigation of Ball Speed Variation

Under lubricated contact, BSV does not loom as a significant problem. However, when lubricant starvation occurs, BSV can be an instigator of retainer distress. The worst case for BSV is under the CMG slewing load, which produces a radial load of about 70 lb ($\approx 3^\circ/\text{sec}$ slew) for an assumed axial load of 40 lb. The discrepancies between Barish and BCL on this subject have been clarified and we are in essential agreement on this subject.

Task IV. Analyses of Separator Stability

Analysis of bearing retainer motions was performed by use of a bearing-dynamics computer program (BASDAP). For the cases run, the results show that the retainer operates in a stable mode. Estimates made on the basis of a retainer stability criterion indicated that unstable motions may arise if the ball-race lubricant film thicknesses are less than 6 μin . This magnitude of film thickness may result under starved lubrication conditions. The computer data also indicated that poor lubrication conditions at the ball-retainer and retainer-race contacts can cause a large change in bearing torque. Hence, such a condition may be manifested in the form of torque glitches as observed from the telemetry data.

FAILURE SCENARIO

The scenario proposed by BCL of the distress of CMG No. 2 and the failure of CMG No. 1 is as follows.

The bearing was operating essentially on the initial charge of lubricant and any additional lubricant that had migrated into the contact region from the lube-nut during ground testing. During space operation of the bearing little, if any, lubricant was fed into the bearing as a result of the following.

- (1) The lube feed system directed the lubricant to the bearing shoulder rather than into the ball-race contact zone, and
- (2) The lubricant could not migrate into the contact region due to
 - (a) An unfavorable thermal gradient
 - (b) Synthane debris and transfer films in the vicinity of the wear track.

As a result of the combination of factors in the lube supply system, the bearing eventually starved as a result of squeeze-film leakage, oil migration, and, possibly, evaporation into the bearing cavity.

As the lubrication condition deteriorated, secondary factors such as BSV under slew load produced wear debris which

- (1) Modified the bearing torque and temperature.
- (2) Further aggravated the migration problem due both to the unfavorable thermal gradient and to inadequate lubricant wetting of the ball-track region.

The torque-temperature glitches abated under conditions of increased external temperature as a result of a shift of the hot region from the ball-race contact region to the bearing housing (i.e., temporary reduction of the axial gradient). This thermal shift produced condensation of the lubricant and favorable lubricant migration. The increased temperature also reduced the propensity of retainer instability and associated torque glitches. The glitches also abated simply as a result of disintegration and dispersion of wear debris.

The remedial action of increasing the bearing cavity temperature and controlling the CMG slew rates doubtlessly enhanced the life of the bearing of CMG No. 2.

INTRODUCTION

On November 23, 1973, Skylab CMG No. 1 incurred a terminal failure. Careful analysis of the available telemetry data indicated that the most probable cause of this failure was lubricant starvation of at least one of the support bearings. At about the same time as the CMG No. 1 failure, the precursors for that failure were also in evidence in CMG No. 2. In an effort to salvage CMG No. 2, remedial action was taken based on a lubricant starvation hypothesis. This action included the following.

- (1) Increasing the temperature of the bearing cavity to encourage lubricant circulation within the bearing.
- (2) Minimizing CMG slewing to reduce bearing damage due to a combined Ball Speed Variation (BSV) and lubricant starvation.

Fortunately, CMG No. 2 did satisfactorily survive the remaining portion of the mission and was eventually commanded dormant at the conclusion of the Skylab program. However, in order to design future spacecraft to avoid the problems incurred by CMG's Nos. 1 and 2, research to elucidate the failure hypotheses has been required.

The purpose of the research program reported herein has been to evaluate possible mechanisms and manifestations of lubricant starvation and the probability that the remedial action did in fact extend the bearing performance. Specifically, the following four tasks have been conducted.

- (1) Oil migration studies aimed at evaluating the circulation (creep) of the lubricant in the bearing cavity.
- (2) Labyrinth seal analyses aimed at determining lubricant loss through the seal.
- (3) Ball speed variation evaluations to clarify some minor discrepancies in earlier postulations.
- (4) Retainer dynamics calculations to determine the possible role of retainer instability on torque glitches.

This report summarizes the results of these tasks.

TASK I. LUBRICANT MIGRATION EVALUATIONS

Lubricant creep and wetting phenomena have long been of interest to the gyro designer. Considerable technology has been accumulated with regard to creep as a depletion and redistribution mechanism in high-speed ball bearings. Most of this work, conducted at the Naval Research⁽¹⁾ and C. S. Draper Laboratories⁽²⁾, has been devoted to the creep of oils away from a bearing (NRL) and the wettability of new, clean bearing surfaces (C. S. Draper Lab.). The effects of debris and transfer films from non-metallic retainer has been largely neglected. Since it appeared that under certain circumstances, the Skylab CMG oiler could deposit lubricant on the bearing shoulder rather than into the ball track area, it has been vital to evaluate the transport of lubricant by creep, or surface migration, from the shoulder area into the ball track. In addition, we evaluated the creep of oil in the presence of a modest thermal gradient in order to determine if the possible gradient between ball track and shoulder would seriously affect oil replenishment by surface migration.

Estimation of Axial Temperature Gradients

The purpose of this analysis is to estimate the axial temperature gradient in the bearing. It is assumed that continuous heating occurs at the ball-race contact zone as a result of ball-spin friction and that this heat is conducted outward through a "semi infinite" steel medium. The magnitude of the heat generation is assumed to equal the mechanical energy dissipation associated with bearing torque. The coordinate system for the analysis is given in Figure 1.

The two-dimensional (conduction) heat flow equation can be written

$$\nabla^2 T(x,y) = 0 \quad (1)$$

where T is temperature. The assumed boundary conditions are

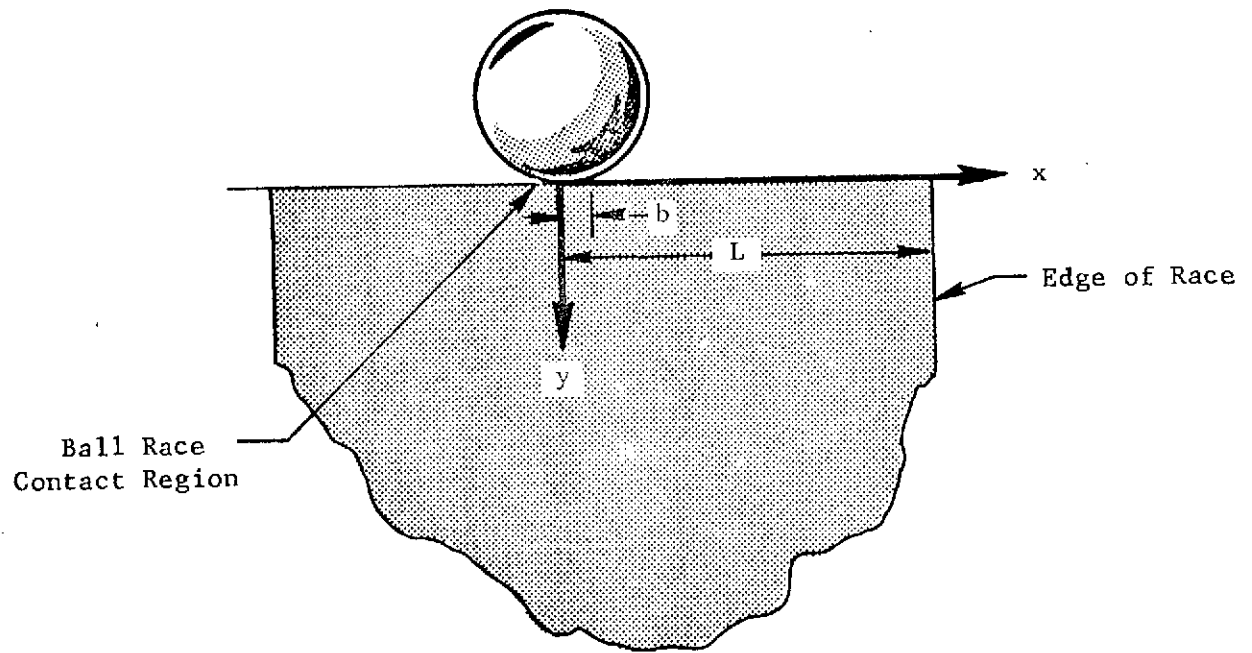


FIGURE 1. COORDINATE SYSTEM FOR AXIAL THERMAL ANALYSIS

$$T(x, \infty) = 0, T_x(0, y) = T_x(L, y) = 0 \quad (2)$$

and

$$T_y(x, 0) = \begin{cases} -\frac{Q}{k} & \text{for } |x| < 1 \\ 0 & \text{for } |x| > 1 \end{cases}$$

where Q is the heat flow per unit area into the race-contact region and K is the conductivity of the steel.

If it is assumed that $\frac{b}{L} \ll 1$ then an approximate solution can be written

$$T = \frac{Q}{K} L \theta$$

where

$$\theta = \sum_{n=1}^{\infty} \frac{2}{n^2 \pi^2} \sin\left(\frac{n\pi b}{L}\right) \cos\left(\frac{n\pi x}{L}\right) \exp\left(\frac{-n\pi y}{L}\right)$$

Values for θ as a function of x are given in Figure 2 for $y = 0$. It should be noted that this solution is only an approximation and that the negative temperature predictions are an artifact of the analysis. The level of the approximation is illustrated by the comparison between the actual and the approximate thermal gradients.

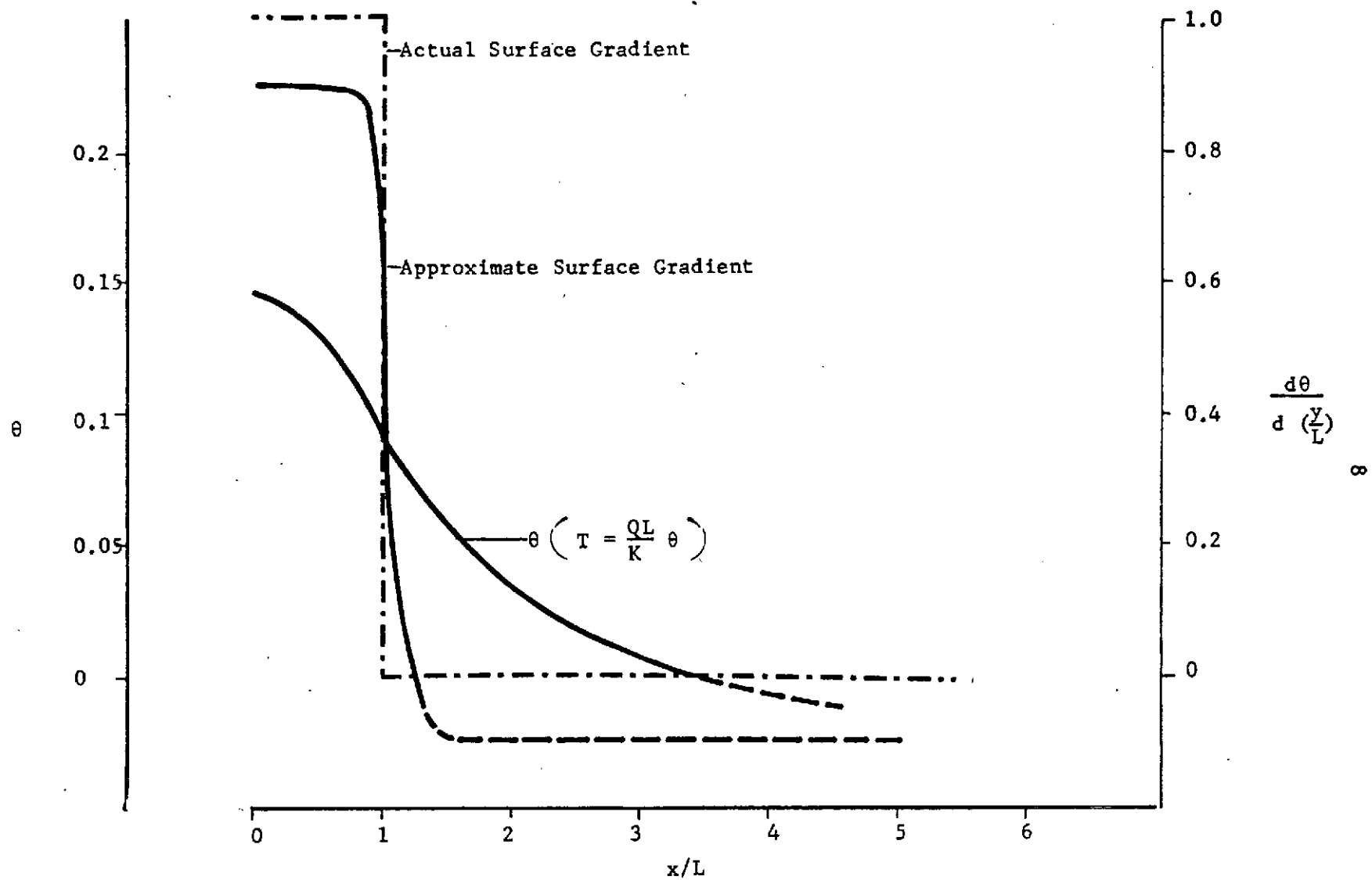


FIGURE 2. APPROXIMATE SOLUTIONS FOR AXIAL THERMAL GRADIENT ANALYSIS

Numerical Example:

Assume:

$$q \approx \frac{\text{Torque}}{2} \cdot \frac{\text{rev}}{\text{sec}} \cdot 2\pi \approx 0.5 \text{ (in.-lb)}(150)(\pi) = 235 \text{ in.-lb/sec}$$

$$A \approx \text{Ball track area} \approx 10 \text{ b (in.)}$$

$$K = 5.4 \text{ lb/sec F}$$

$$L = 10 \text{ b}$$

$$\frac{QL}{K} = \frac{qL}{AK} = 43.5$$

Then

$$\Delta T_{\max} = \theta_{\max} \frac{QL}{K} = (0.14)(43.5) = 6.1 \text{ F}$$

Thus, for a torque of 0.5 in.-lb, an axial thermal gradient on the order of 6 F is possible. Realistically, this gradient is most likely on the order of say 2 to 10 F.

Experimental Method

Two general types of vacuum creep tests were carried out in the apparatus shown in Figure 3. All tests were run at a pressure of 10^{-5} torr. A liquid nitrogen trap was used to prevent back-streaming of pump oil.

To determine the effects on creep of synthane transfer films and wear debris, the highly-polished circular disk shown in Figure 4 was employed. The disk was prewetted with KG-80 and the excess was wiped-off after setting over night. This rest period with a relatively thick oil film was used as a wettability test for the surface of the plate. The excess oil was then wiped-off and the disk installed on the spindle in the vacuum chamber. A



FIGURE 3. OVERALL VIEW OF CREEP EXPERIMENTAL APPARATUS

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OF POOR QUALITY

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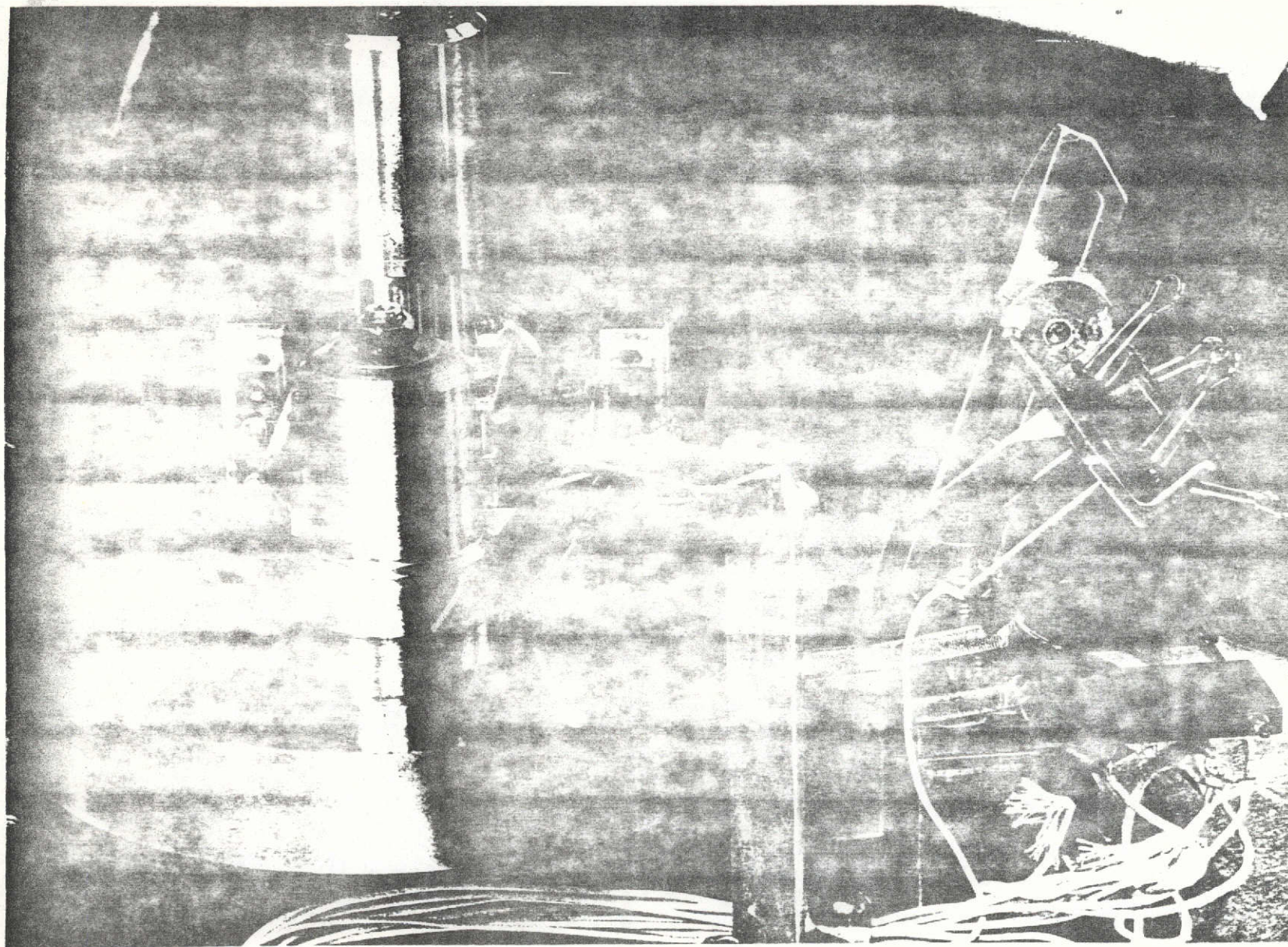


FIGURE 4. DETAIL OF THE CIRCULAR PLATE SPECIMEN USED IN TRANSFER FILM
WETTING EXPERIMENTS (CENTER) AND THE PLATES USED IN THE
THERMAL GRADIENT EXPERIMENTS (RIGHT)

block of synthane vacuum impregnated with KG-80 was installed in the rider shown just behind the spindle in Figure 4. A 500-gram dead-weight load was applied and the disk was rotated at 2000 rpm for 12-16 hours in vacuum. This procedure produced a transfer film that was barely discernible with the naked eye. The system was opened and spots of KG-80, SRG-60, and SRG-160 were placed on the disk such that in creeping outward the oils would intercept the synthane transfer film and creep inward could proceed uninhibited. The system was repumped to $\sim 10^{-5}$ torr and the position of the top edge of the drops, (the edge nearest the spindle and farthest from the transfer film) and bottom edge (the edge nearest the transfer film) determined using a system of mirrors (Figure 4) and a precision cathetometer. Position readings were taken twice daily for over 400 hours.

The thermal gradient experiment was carried out using the two polished plates shown to the right of the disk assembly in Figure 4. Two identical polished plates were fastened on one end to a massive heat sink attached to the vacuum chamber base plate. A small resistance heater was installed in the other end. This arrangement made it possible to achieve and sustain a temperature gradient of 4 to 5 F between the two ends of both plates. The plates were coated with KG-80, checked for wettability and wiped in the same manner as previously described in connection with the large disk. A spot of KG-80 was placed on each plate--on the hot end of one and the cold end of the other. Baseline spot position readings were taken using the cathetometer. The system was pumped for ~ 24 hours before the heater was turned on to produce conditions of 87 F on the cold end and 91 F on the hot end. Migration readings in the form of drop edge (top and bottom of each drop) positions were taken twice daily.

Results

Figures 5 and 6 summarize the results of the experiments to determine the oil migration in the presence of a transfer film. The data from two replicate experiments with KG-80 are summarized in Figure 5 in terms of the position of the top of each spot, and the bottom of each spot relative to a reference point as a function of time in vacuum. Recall that the top edge of each drop is that edge that is away from the transfer film track and is creeping more or less unimpeded over the surface prewetted with

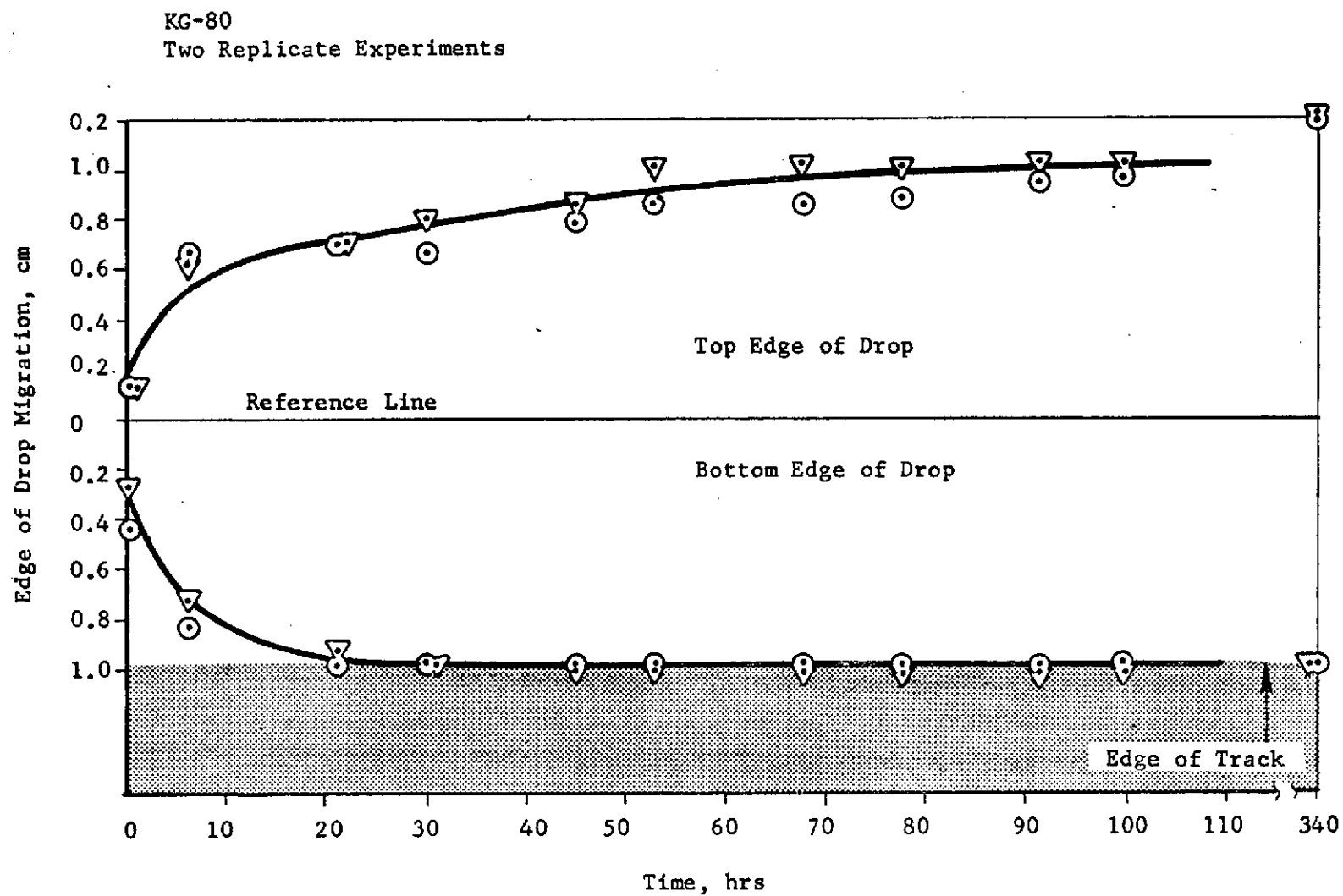


FIGURE 5. RESULTS OF CREEP EXPERIMENTS IN THE PRESENCE OF A SYNTHANE TRANSFER FILM - KG-80 OIL

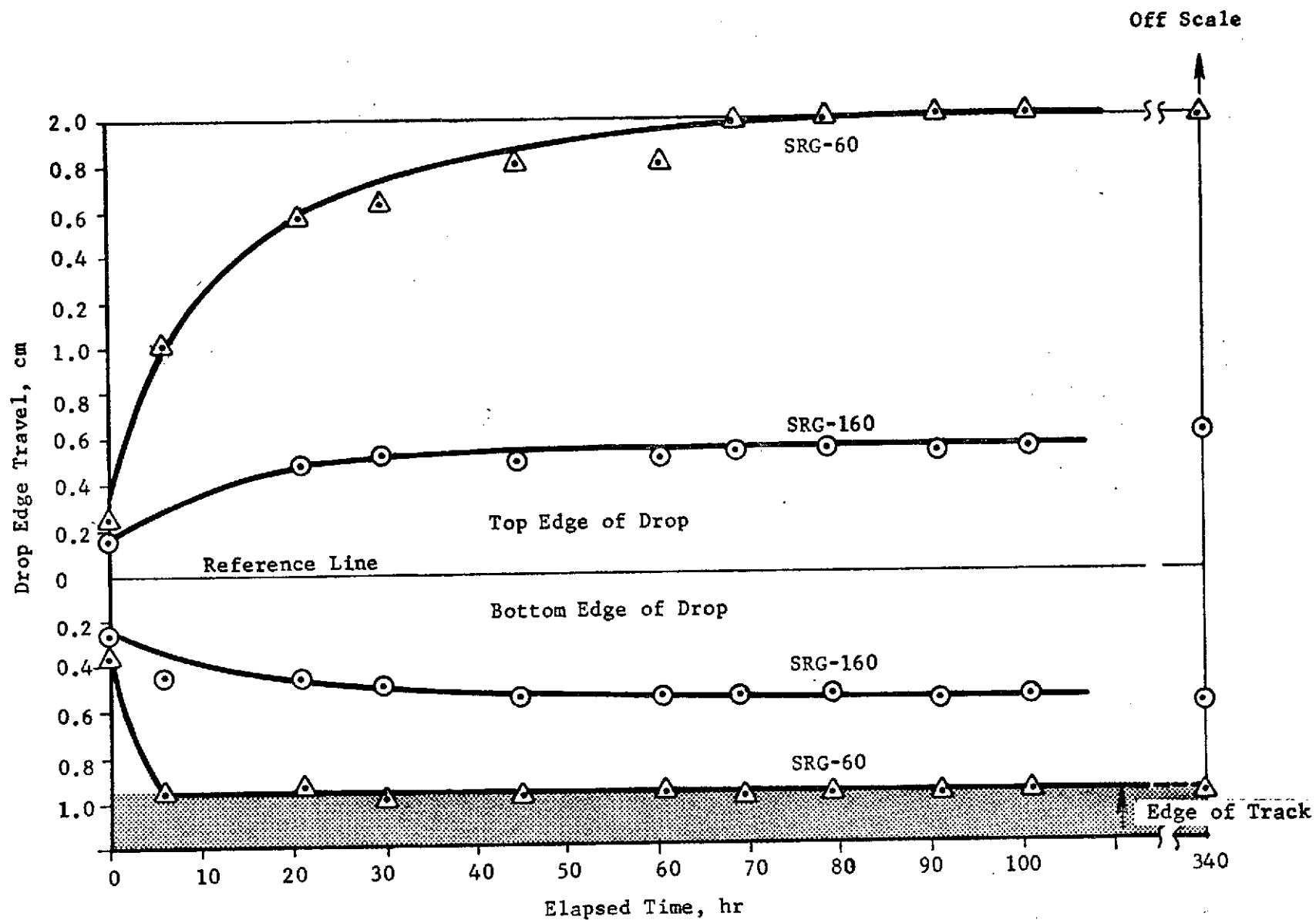


FIGURE 6. RESULTS OF CREEP EXPERIMENTS IN THE PRESENCE OF A SYNTHANE TRANSFER FILM - SRG-60 AND SRG-160 OILS

KG-80. Note that the bottom edge of the drop intercepted the edge of the track within 20 hours and did not cross this barrier in the 340 hours of the experiment. (The rather rapid migration rate for the first 20 hours is probably due to spreading rather than creep and might not be expected to occur in zero g.) As seen in Figure 4, the sister oil, SRG-60, was likewise impeded by the transfer film. The SRG-160 did not reach the critical region during the test. After 40-50 hours it was noted that the KG-80 from the prewetting in the transfer film track had beaded up, indicating a severe nonwetting condition brought about by a very thin transfer film of synthane.

A separate, subjective experiment was carried out wherein the block of synthane, was replaced by a ball riding in a impregnated synthane pocket which was dead-weight loaded to 600 grams. Rotation of this assembly on the disk surface produced a credible appearing ball track having a narrow transfer film band flanked by some debris build up. None of the oils would cross this barrier in ~400 hours of vacuum testing. A posttest Talysurf revealed no wear of the steel plate. Hence, the barrier consisted only of synthane debris and transfer film. Other cursory tests indicate that porous polyimide likewise produces nonwetttable surface films and debris barriers.

The results of the thermal gradient experiments are shown in Figure 7, 8, and 9. In Figures 7 and 9 the drop migration is expressed as the position of the top and bottom of the drops as a function of time. Figure 7 shows the motion of the drop initially placed on the hot end of one of the plates. Figure 8 shows a sketch of the drop migration in the course of the 340-hour experiment. Note that the drop migrated to the cold end of the plate, leaving only a residual film on the hot end (bottom curve in Figure 7). The drop placed on the cold end of the other plate showed only 1.6 cm of migration at 87 F (bottom curve; the top of the drop went off the plate)--a figure that compares favorably to the 1.2 cm migration observed for KG-80 at 75 F in the transfer film experiment.

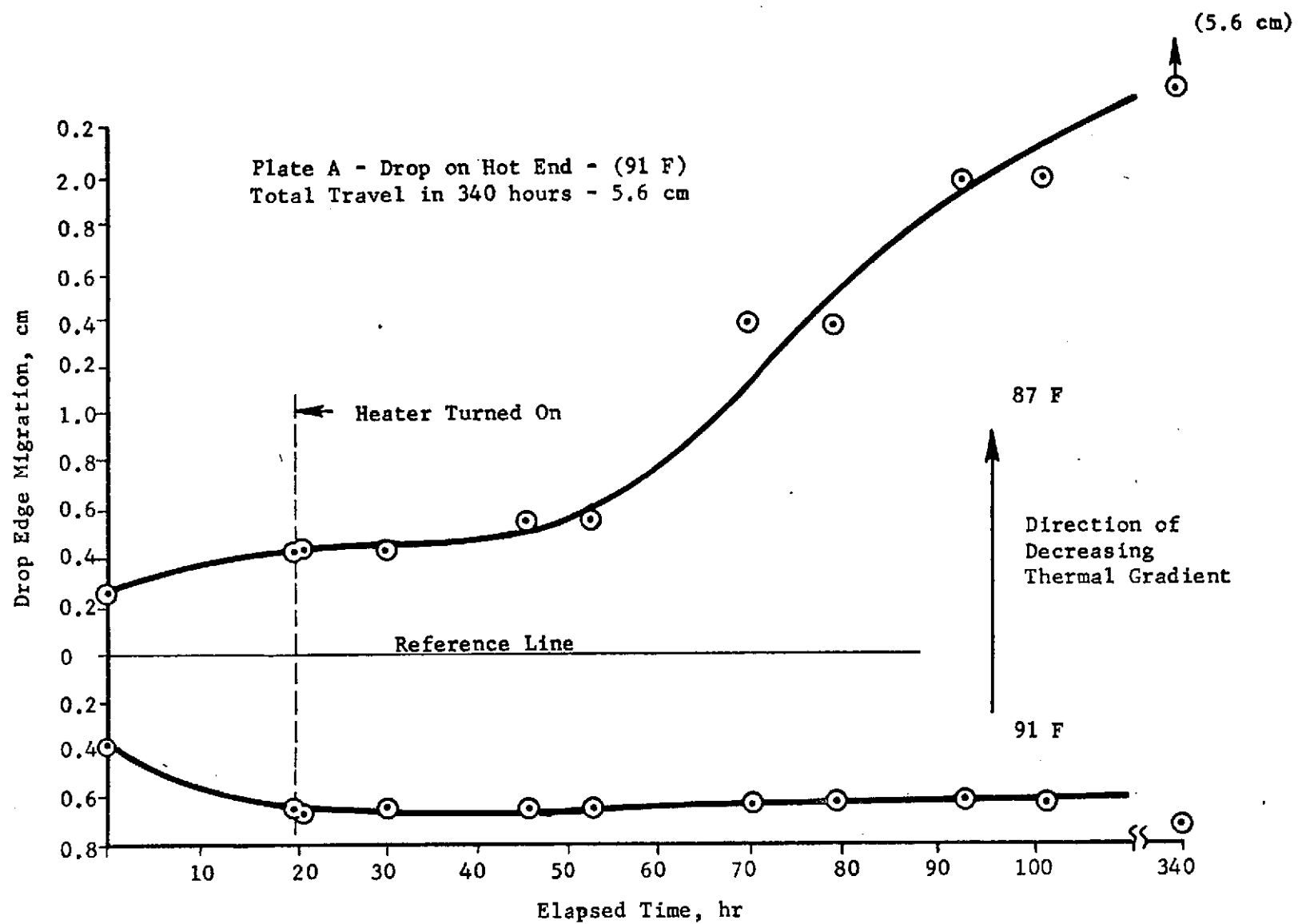


FIGURE 7. RESULTS OF THE THERMAL GRADIENT CREEP EXPERIMENTS - KG-80 ON HOT END

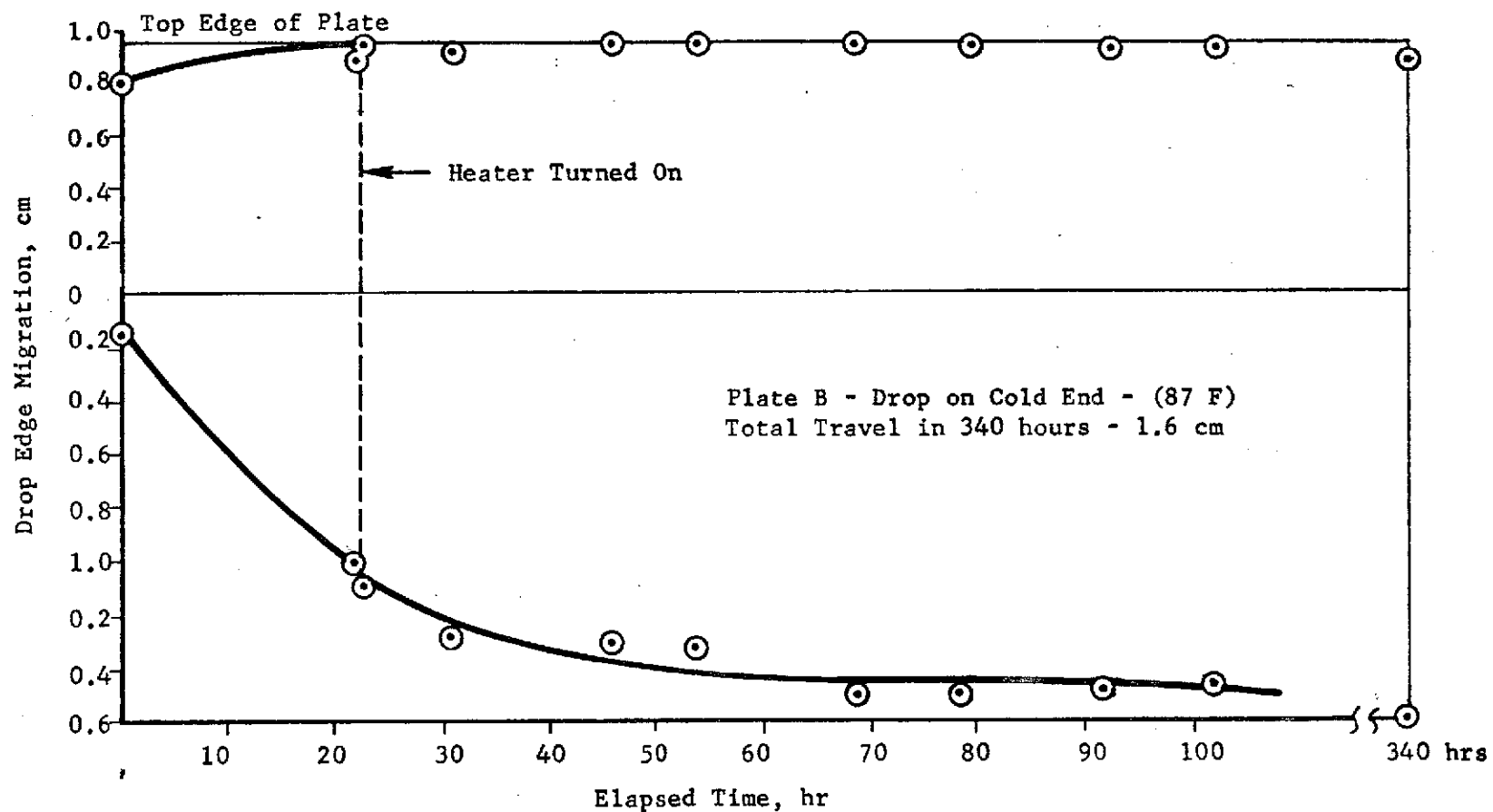


FIGURE 8. RESULTS OF THE THERMAL GRADIENT CREEP EXPERIMENTS - KG-80 ON COLD END

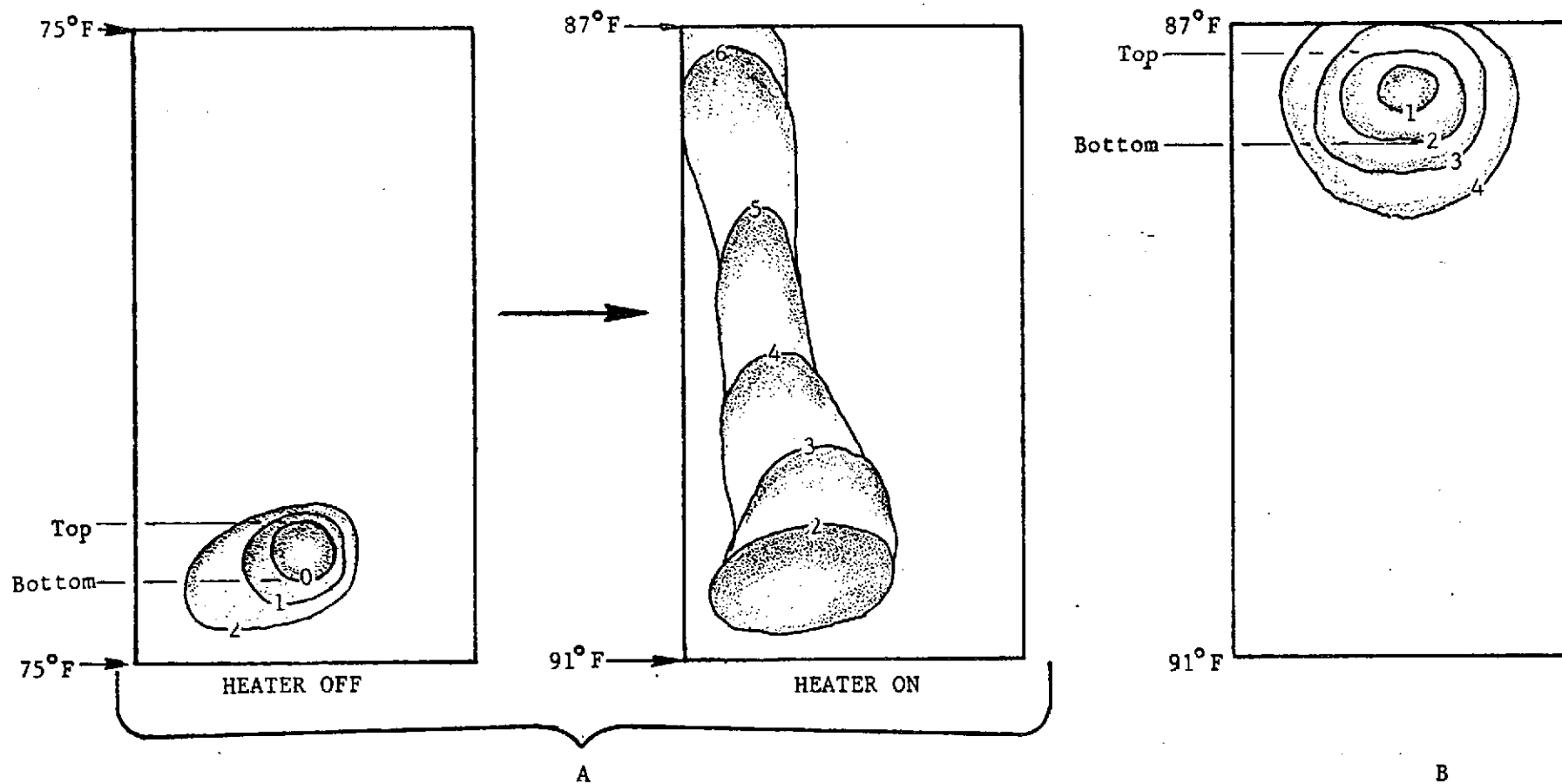


FIGURE 9. SKETCH SHOWING MIGRATION PATTERNS OF KG-80 IN THE PRESENCE OF A 4 F THERMAL GRADIENT

Evaluation of Lubricant Loss Due to Centrifugal Force

One additional possibility for lubricant loss which has been postulated by Barish⁽³⁾ is associated with the centrifugal-slinging of the lubricant on the balls and races. Since the bearing rotates at moderately high speeds (~9,000 rpm) this possibility does loom as a mechanism for lubricant starvation. However, there are a considerable number of bearings operating at speeds significantly higher than 9,000 rpm that do not incur starvation failure. Nonetheless, the unique lubricant circulatory system used in the CMG poses new questions pertaining to role of centrifugal effects.

To evaluate lubricant loss due to centrifugal action, some cursory experiments have been conducted using the BCL rolling-disk apparatus. This apparatus consists of two counter-rotating disks which can be loaded together into lubricated contact (Figure 10). The apparatus is equipped with various types of instrumentation including a device for monitoring the extent of lubrication using an electrical conductivity technique as shown in the Figure. The disks in the apparatus are 1.42-inch in diameter and can be rotated at speeds to 20,000 rpm. For the experiment reported here, the contact load was set so as to produce a maximum Hertz contact stress of 150,000 psi. The instrumentation allows for an assessment of the percentage of lubrication in this contact. (I.e., 0 percent implies a dead electrical short across the disks where 100 percent implies an open circuit.)

The sequence for the centrifugal starvation experiments was as follows.

- (1) The disks were solvent cleaned and were coated with a minimum of 150 mg of KG-80 lubricant.
- (2) The disks were rotated at the desired speed for 1 minute out of contact to sling off the lubricant.
- (3) The disks were brought into contact and the extent of lubricant film was measured. This was invariably (initially) 100 percent film.

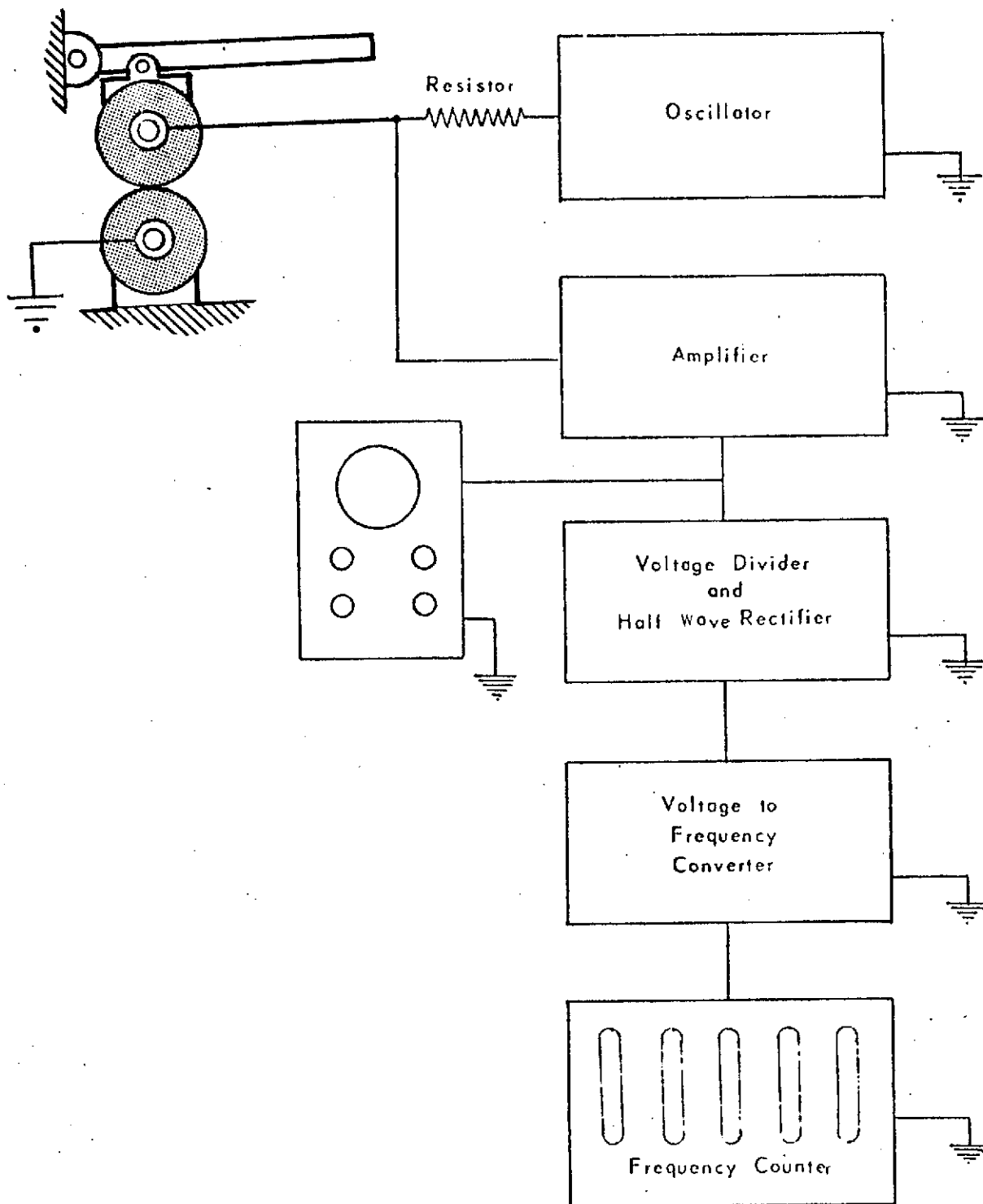


FIGURE 10. BLOCK DIAGRAM FOR CONTACT CONDUCTIVITY EXPERIMENTS

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- (4) The disks were rotated together under load until a 70 percent film indication occurred and a record of the time for this occurrence was recorded.
- (5) The sequence was repeated for other disk speeds. In all, speeds of 5,000, 7,500, and 10,000 rpm were evaluated. These speeds should be indicative of range of rotational speeds in a bearing.

A plot of speed (rpm) versus time for the film to collapse is given in Figure 11. At no time did instantaneous collapse occur. Further, the collapse time is primarily related to the number of times the lubricant passes through contact. That is, lubricant starvation is more related to the squeeze film effect due to contact pressure than due to centrifugal force effect. Any speed phenomena beyond this number of cycles was within the scatter of the data.

Our general conclusion is that centrifugal slinging of the lubricant is not a uniquely significant factor for the CMG bearing. This conclusion is heavily supported by the fact that many life-test units as well as extraterrestrial units have operated successfully under centrifugal slinging conditions that are identical with those experienced by the failed bearing.

Conclusions

On the basis of these simple experiments it is possible to draw several conclusions.

- (1) Nonmetallic retainer materials give rise to oil migration barriers in the area of the ball track. These barriers are (a) transfer films that are not wetted by the lubricant, and (b) debris piles that act as physical barriers.
- (2) Under conditions of operation wherein bearing contact angles are changing due to load changes, this nonwetting area can be somewhat broader than the nominal ball track and can effectively prevent replenishment of oil in the critical ball-track region by surface-migration mechanisms.

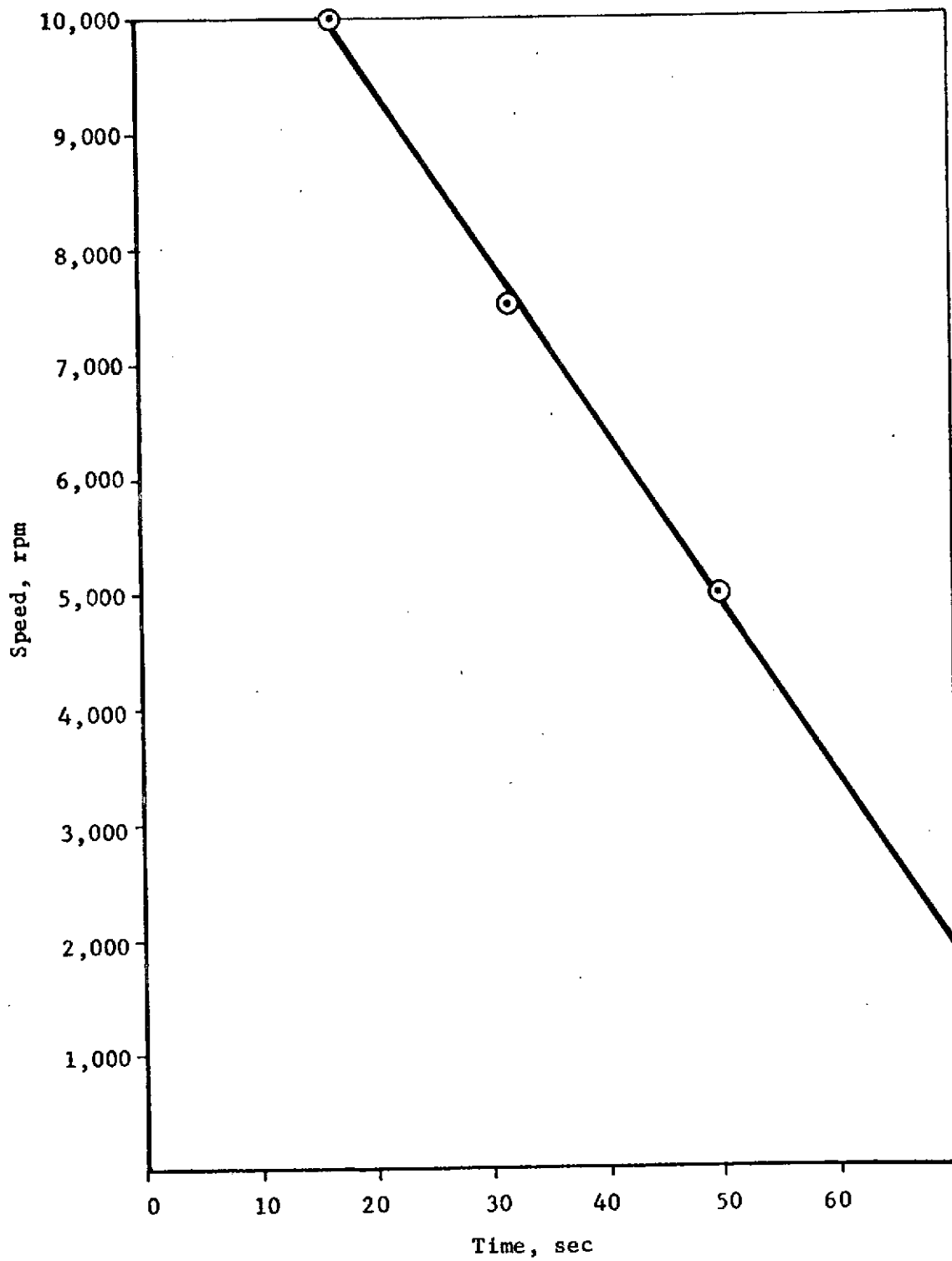


FIGURE 11. TIME TO COLLAPSE OF EHD FILM IN A CENTRIFUGALLY - DEPLETED CONTACT

- (3) Oil tends to creep from hot to cold⁽¹⁾. A gradient as low as 4 F, shown by analysis to be reasonable, can not only prevent oil from creeping into the ball track region, but can also force oil to migrate away from the ball-track region.
- (4) Lubricant loss due to centrifugal force does not appear to represent a significant mechanism for the Skylab CMG.

TASK II. LABYRINTH SEAL EVALUATIONS

The purpose of this task was to analyze lubricant loss rate through the labyrinth seals in order to assess the possibility that such a mechanism could be responsible for lubricant depletion within the bearing leading to bearing failure. The approach taken was to estimate, analytically, the seal loss rate based on classical flow theory and to compare the computed values with the lubricant feed rate to the bearing.

It is assumed that molecular flow is the governing mechanism for flow of lubricant through the labyrinth seals. The supply of lubricant potentially available for loss through the seals will depend on the equilibrium condition established by the fluid within the closed chamber. Here, equilibrium refers to the balance between molecular addition and removal of lubricant to and from various surfaces of the sealed chamber. The balance is a function of the vapor pressure and temperature of the fluid which may vary throughout the chamber. The estimates of lubricant loss rate based on the foregoing assumptions are summarized below. In the determination of the estimate, the assignment of parameter values has been conservative so that lubricant-loss predictions are believed to be somewhat higher than those which would actually obtain.

Molecular Flow Analysis

Vapor Pressure of KG-80

The driving force for molecular (mass) flow through the labyrinth is the number and energy level of the molecules of lubricant on the bearing-chamber side of the annulus (labyrinth). In turn, the number and energy level are dependent on the state of the lubricant at the bearing surfaces and at any other of the inner chamber surfaces at which lubricant has collected. Initially, the potential source for loss through the seals is small; however, due to creep and to vaporization from the bearing areas and subsequent condensation on the surrounding walls, an equilibrium condition will be established wherein the pressure of the lubricant vapor in the

bearing-chamber side will be the vapor pressure of the lubricant at the temperature of the lubricant.

Thus, the knowledge of the vapor pressure of the lubricant as a function of temperature is a critical parameter for estimating lubricant loss through the seals. Unfortunately, very little data concerning KG-80 are available for the conditions of interest here. Vapor-pressure data for KG-80 have been estimated by extrapolation of data from other studies^(4,5). The results of this extrapolation are shown in Figure 12. For example, it is estimated that the vapor pressure of KG-80 at 100 F is 1×10^{-7} torr. This value agrees with that quoted by Singer at MIT*, and hence, is taken to be representative of KG-80. Of note from Figure 12 is the significant change in vapor pressure over a very short range of temperature. Also, concerning the question of the assumption of molecular-flow regime, Apt and Wiederhorn⁽⁶⁾ calculated a mean-free path of 300 cm (118 in.) for molecules of an oil having a vapor pressure of 10^{-6} torr at a temperature of 290 K (63 F). Based on this, the mean-free path for molecules in the present situation should be somewhat greater; consequently, the criterion for the molecular-flow regime--a mean-free path that is greater than any linear dimension within the chamber--would appear to be justified.

Estimate of Equilibrium Condition within the Sealed Chamber

In molecular flow, the number of molecules that strike a unit area per unit time is given by

$$v_n = n \frac{\bar{C}}{4} \quad , \quad (3)$$

where n is the number of molecules of gas per cm^3 and \bar{C} is the arithmetical average velocity of those molecules. \bar{C} is given by

$$\bar{C} = \sqrt{\frac{8RT}{\pi M}} \quad . \quad (4)$$

Here, M is the molecular weight of the gas, R is the gas constant (8.31×10^7 ergs/g-mole/K), and T is the absolute temperature. Substitution

*Personal communication.

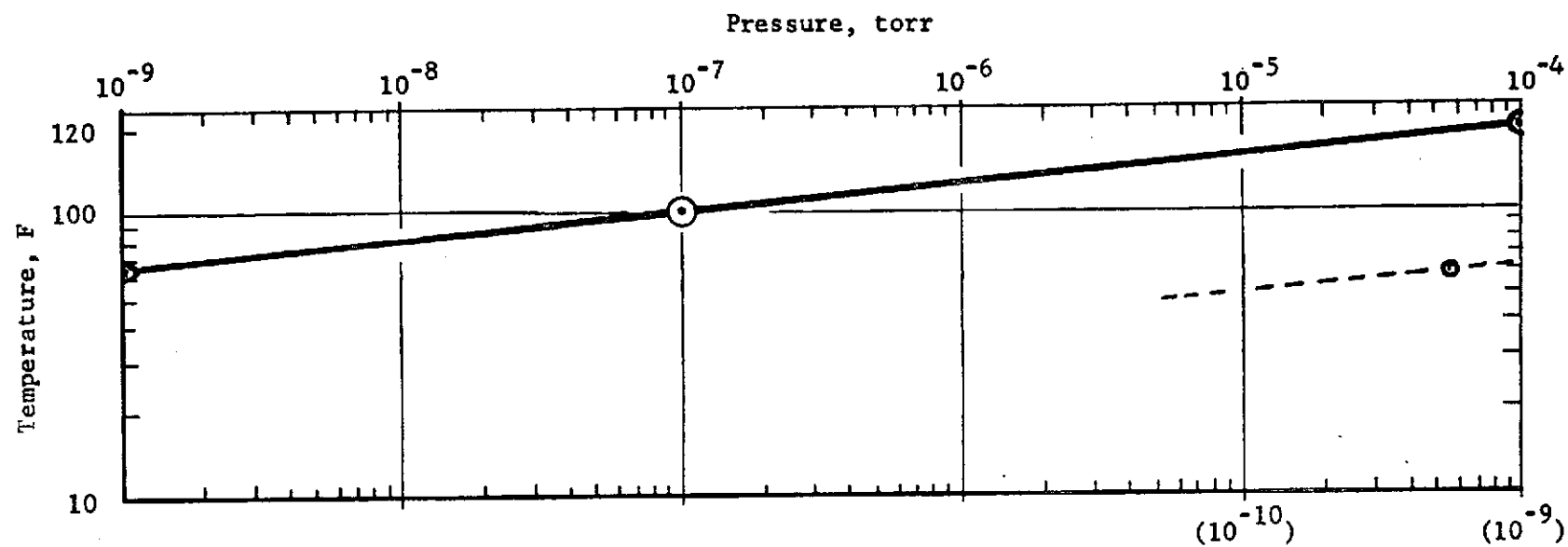


FIGURE 12. VAPOR PRESSURE OF KG-80

of Equation 4 into Equation 3 and use of the gas law yields

$$v_n = \frac{Np}{(2\pi RTM)^{\frac{1}{2}}} \quad , \quad (5)$$

where p is the pressure and N is the total number of molecules. Equation 5 is multiplied by M/N , the mass per molecule, to obtain the total mass of material (in grams) per second striking a unit area of 1 cm^2 , i.e.,

$$v_g = p \left(\frac{M}{2\pi RT} \right)^{\frac{1}{2}} = 1.60 \times 10^{-3} p \left(\frac{M}{P} \right)^{\frac{1}{2}} \quad . \quad (6)$$

Here it is assumed that KG-80 has a molecular weight of 598*, and the operating temperature is 100 F or $\sim 310 \text{ K}$. Applying these values to Equation 6 there results

$$v_g = 2.22 \times 10^{-10} \text{ g/cm}^2/\text{sec} \quad . \quad (7)$$

The result shown in Equation 7 indicates the evaporation rate of the lubricant at 100 F. If the vapor contained in the sealed region of the CMG is in equilibrium and if all of the surfaces within the sealed region are at uniform temperature, then Equation 7 also represents the rate of condensation. Hence, the net transfer from any surface is zero. However, a temperature gradient exists between the bearing surfaces and the walls of the enclosed chamber. The critical areas for lubricant loss within the bearings are those of the races and balls. It is estimated that this area amounts to 50 cm^2 . If the surrounding areas of the enclosure are at 60 F the net loss rate from the bearing based on an exposed area of 50 cm^2 could be rather significant (of the order of $40 \text{ } \mu\text{g/hr}$) compared to the supply

*H. Singer, MIT.

rate to the bearing. However, lubricant depletion from the bearing-race region by molecular flow is impeded by the annuli formed by the bearing races and the cage. The total area of the annuli on both sides of the bearing is estimated to be 2 cm^2 . On this basis, the loss rate to the surrounding enclosure at 60 F from the bearing-race region at 100 F is estimated to be approximately $1.6 \text{ } \mu\text{g/hr}$. Estimates of the net loss from the bearing-race region to the surrounding walls of the enclosure at 60 F have been made for a range of bearing surface temperatures. The results are shown in Figure 13. It is noted that the net loss rates by molecular flow are small compared to the supply rate to the bearing.

Estimate of Lubricant Loss Through Labyrinth Seals

Equation 7 is taken as the basis for calculating the amount of lubricant per unit area striking the inner wall in the vicinity of the labyrinth seals. This is an overestimate inasmuch as it is expected that the temperature of the lubricant in the vicinity of the seal annulus will be at a lower temperature than the 100 F level on which Equation 7 is based. The annular shaft-clearance gap, an integral part of the labyrinth seal configuration, may be regarded as removal of an area of the wall. To determine the flow of lubricant vapor through the gap, account is also taken of the length of the clearance. This produces an additional reduction to molecular gas flow.

This area of the annular gap clearance may be estimated by application of the formula

$$A = 2.03 \times 10^{-2} DC \quad (8)$$

where D is the shaft OD (inches), C is the seal clearance (mils), and A is the area (cm^2). From drawings of the CMG unit the dimensions involved are

$$D = 1.812 - 0.001 \text{ in.}$$

$$\text{ID of seal} = 1.843 + 0.003 \text{ in.}$$

Clearance of seal gap is therefore

$$C = \sim 18 \text{ mils (max)}$$

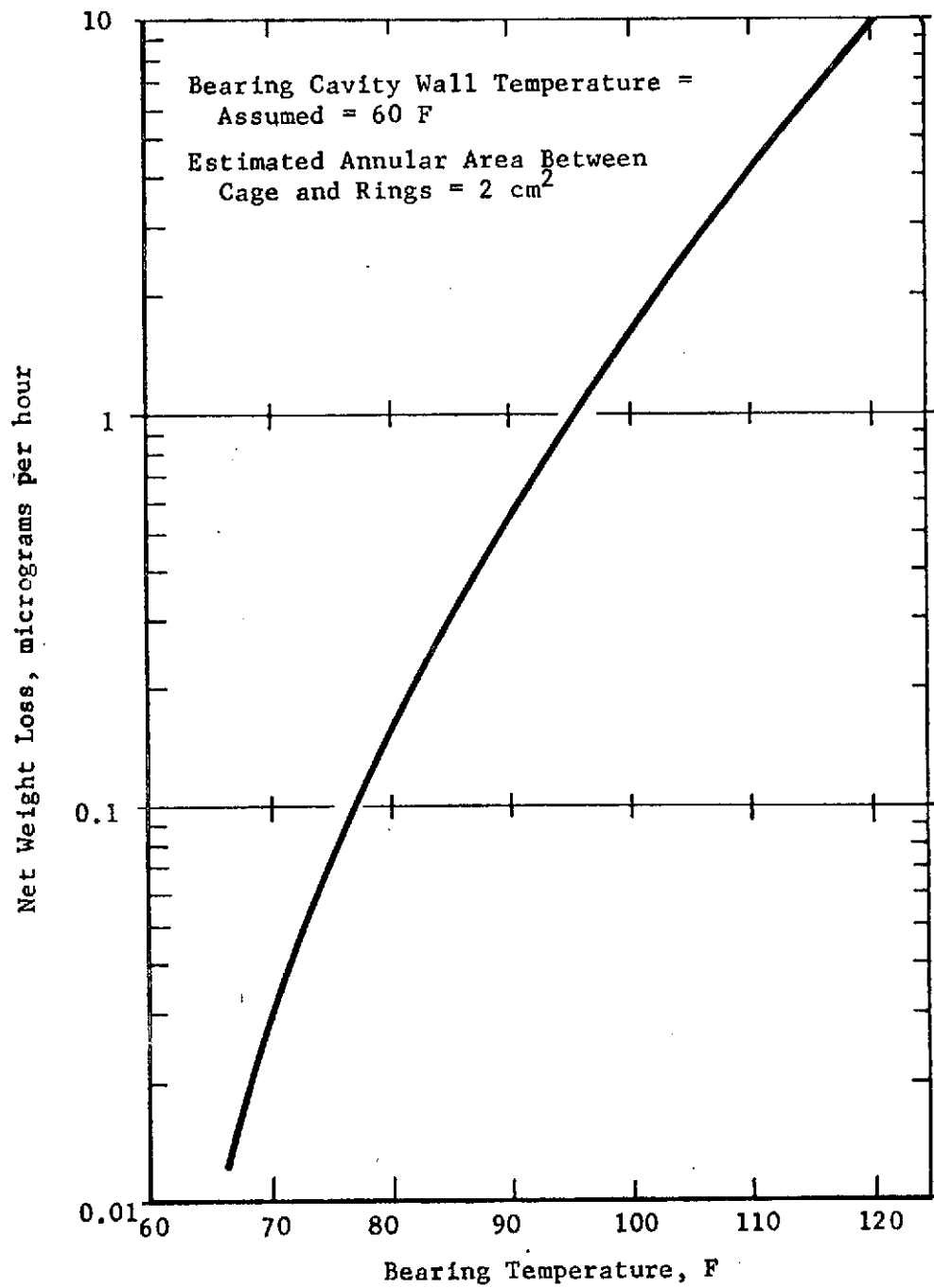


FIGURE 13 KG-80 LUBRICANT LOSS FROM BALL-RACE AREA OF CMG BEARING (SIZE 107)

and

$$A = 0.662 \text{ cm}^2$$

The correction to be made to molecular flow rate through the annulus is defined by the following:

$$K = \left(L + \frac{3}{8} \frac{t}{d} \right)^{-1} \quad (9)$$

where t/d is the ratio of the thickness of the aperture to its diameter. The factor is applied as a modification of Equation 6 as follows:

$$v_g = 1.60 \times 10^{-3} \text{ pK} \left(\frac{M}{T} \right)^{\frac{1}{2}} \quad (10)$$

The labyrinth seal is composed of seven straight segments, actually annular lands, and five annular grooved segments--each having a semicircular cross section of 0.031-inch radius. These annular grooves provide expanded "pocket" spaces which effectively "turn back" a sizeable fraction of the escaping lubricant molecules. For simplicity, this special geometry will be disregarded in the first approach to estimating seal leakage. For this consideration the clearance gap will be treated as a straight-walled tube of 0.765-inch length. Also, the t/d ratio for this simplified geometry is

$$\frac{t}{d} = 0.415$$

By applying the annular clearance area and the value for the annulus length correction, Equation 9, to the lubricant loss estimate, Equation 7, we obtain

$$v_g = \sim 0.46 \text{ } \mu\text{g/hr}$$

The effect of seal clearance on loss through the labyrinth seal based on the above considerations is shown in Figure 14. The loss rate is estimated to be negligible with respect to the rate supplied to the bearing.

Another method⁽⁴⁾ of estimating loss rate through labyrinth seals is based on gas volume considerations. General conductance is defined as

$$F = K \sqrt{\frac{T}{M}} \quad , \quad (11)$$

where

F = Conductance, liters/sec

M = Mass of gas molecules, amu.

K = Area factor depending on geometry.

For several conductances in series, the overall conductance is

$$F_T = \frac{1}{\frac{1}{F_1} + \frac{1}{F_2} + \frac{1}{F_3} \dots} \quad . \quad (12)$$

This treatment further separates the entrance conductance, F_o , from the length consideration, F_a . The entrance formula is

$$F_o = 3.63 A \sqrt{\frac{T}{M}} \quad (13)$$

where A = Area of entrance in cm^2 . The formula given for the annulus length is

$$F_a = 9.7\pi (a_1 - a_2)^2 (a_1 + a_2) \frac{1}{\ell} \sqrt{\frac{T}{M}} \quad (14)$$

where

a_1 = Radius of outer cylinder, cm

a_2 = Radius of inner cylinder, cm .

For a shaft of 1.812 in. - 0.001 diameter and a housing of 1.843 in. + 0.003 diameter.

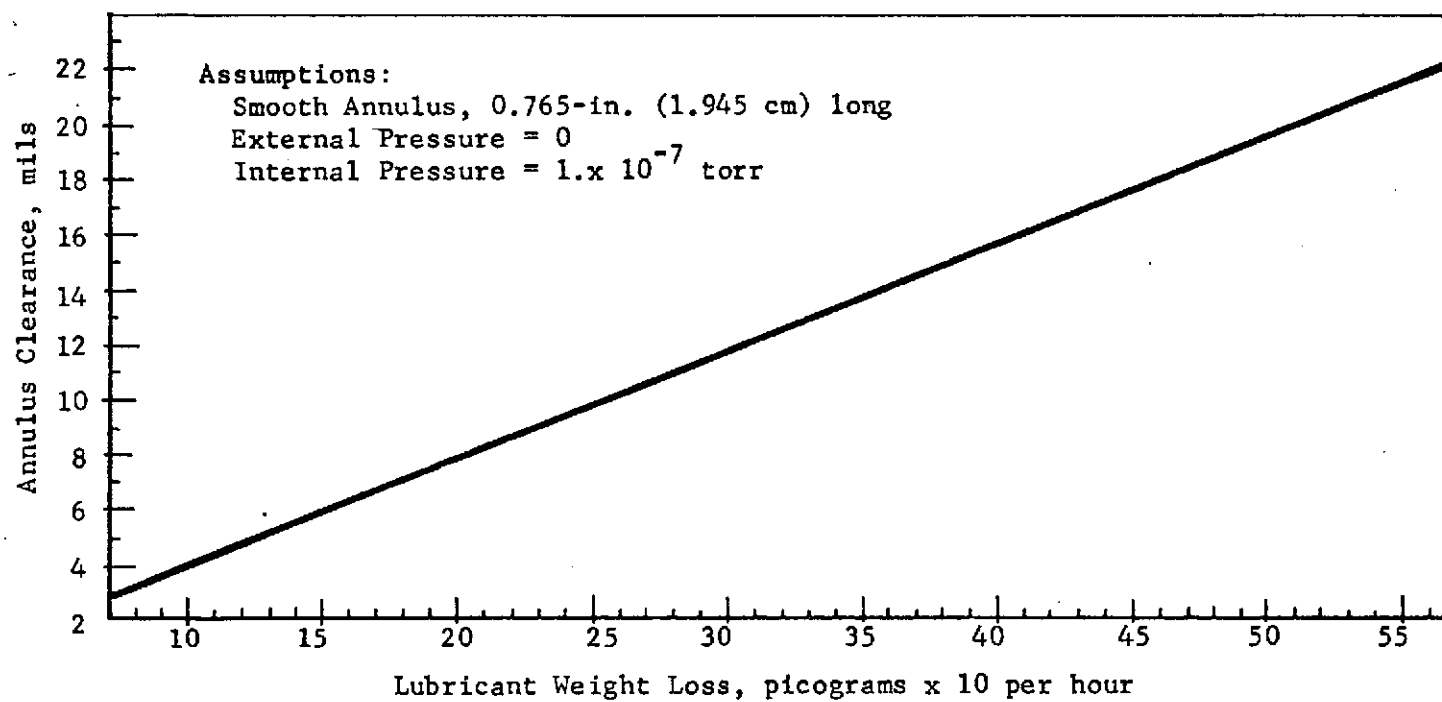


FIGURE 14. KG-80 LUBRICANT LOSS BY DIFFUSION THROUGH SHAFT LABYRINTH SEAL FOR TEMPERATURE OF 100 F (310 K)

$$a_1 + a_2 = 4.64 \text{ cm} .$$

Also, $l = 0.765 \text{ inches (1.945 cm)}$
 $T = \sim 310 \text{ K}$
 $M = 598 \text{ amu}.$

Substitution of these values in Equation 14 yields:

$$F_a = 8.38 \times 10^{-2} \text{ liters/sec} .$$

Also, from Equation 13

$$FF_o = 3.63 A (0.518)^{\frac{1}{2}} .$$

With 0.662 cm^2

$$F_o = 1.73 \text{ liters/sec} .$$

Finally F_T is estimated by applying Equation 12

$$F_T = \frac{1}{12.48} = \sim 0.08 \text{ liters/sec} . \quad (15)$$

To convert to mass flow rate,

$$Q = pF \quad (16)$$

where

Q = Quantity, torr liters/sec
 p = Differential pressure, torr
 F = Conductance, liters/sec.

In terms of mass,

$$G = \frac{pFM}{1.7 \times 10^4} . \quad (17)$$

By substituting the values found above it is calculated that

$$G = \sim 1 \mu\text{g/hr} \quad . \quad (18)$$

This value for mass-flow conductance is about two times as large as the rate calculated by the previous method. In any case the loss rate estimated above is negligible when compared with the supply rate to the bearing.

Conclusion

Lubricant loss rate through the labyrinth seal has been estimated analytically on the basis of molecular flow. First, vapor pressure data for KG-80 were determined as a function of temperature by extrapolation from existing data on lubricants in the range of interest. Then, the equilibrium condition of the lubricant within the sealed chamber was determined in order to obtain an estimate of the rate of lubricant molecules striking the labyrinth seal annular area on the inlet side. Estimates indicated that at 100 F and 10^{-7} torr vapor pressure, the loss rate through the seal by molecular flow is 0.5 to 1 $\mu\text{g/hr}$. This is an overestimate based on a conservative choice of parameters. Since the predicted loss rate through the seal is small with respect to the supply rate to the bearing (of the order of 40 $\mu\text{g/hr}$)⁽⁷⁾ it is concluded that lubricant depletion by molecular flow is not likely to occur.

TASK III. BALL SPEED VARIATION (BSV) ANALYSES

The spin-axis bearings for the Skylab CMG's represent a reasonable configuration for the application. Under well-lubricated conditions, the bearings should operate indefinitely without failure. However, the bearings do experience rather severe radial/axial load situations which, under poor lubrication conditions, can trigger bearing distress. Specifically, under certain combinations of radial and axial loads considerable variations in the ball speeds can occur; when this speed variation is large, the variation in the gap between the balls can be sufficient to "take up" the pocket clearances. Under extreme BSV, significant ball-pocket loading will occur which (under starved lubrication conditions) can result (hypothetically) in retainer wear or fracture.

Ball-Pocket Load

During the course of the evaluation of the P-11 test failure, as well as the Skylab CMG failure, considerable attention has been given to evaluation of (1) the conditions which promote excessive BSV, and (2) the effect of BSV on retainer distress. Such BSV studies have been conducted both at BCL^(8,9) and by Barish^(3,10,11).

The early conclusion reached in the BCL studies⁽⁸⁾ was that under adequate lubrication conditions BSV could not cause retainer failure. That is, if the ball-race interface contains a complete lubricant (EHD) film, then the ball can slide as well as roll on the lubricant without generating severe forces that would be transmitted to the retainer. Thus, even if severe BSV occurs, the lead and lagging balls would simply slip and accommodate the excessive excursions. Forces of only about 1 lb would be developed even under the very severe BSV condition experienced in the P-11 fixture. Further, these forces would be considerably less in space due to the low level of axial loading. Nevertheless, lubricant starvation occurs when a dry contact situation at the ball-race interfaces could produce severe retainer loading.

Condition Promoting Excessive BSV

Some consensus seemed to evolve in the P-11 failure reports pertaining to the fact that a combination of poor lubrication coupled with BSV would be required to promote bearing distress. However at the demise of Skylab CMG No. 1 some discrepancies appeared to exist between Barish and BCL on the conditions promoting excessive BSV. Before discussing the nature of the discrepancy it is helpful to discuss first the effect of loading on BSV.

Under pure axial load, all the balls in a perfect bearing will have the same contact angle and hence will rotate at the same velocity. Under a combined radial and axial load, however, some balls tend towards unloading whereas some tend toward an overload condition. Under this load combination, all balls experience different contact angles; therefore, they rotate at different velocities. That is, the velocity of the ball is directly related to the distance R_i (Figure 15) which is the velocity of the inner race at the point of contact. R_i is, in turn, related to the inner contact angle β_i .

In the BCL report⁽⁹⁾ on the Skylab CMG's, a graph of BSV as a function of radial load was presented. This graph Figure 16 was based on bearing dynamics calculations at BCL of the type discussed in Reference 8. The analyses indicated that radial loads in excess of 70 lb (assuming an axial preload of 40 lb) could produce excessive BSV which could lead to bearing distress under poor lubrication conditions. The 70-lb load represents a slew rate of nominally 3°/sec. The graph was intended to serve as a guideline to develop policy to limit CMG slew to the 3°/sec level. A policy to limit slew rate was adopted and this policy along with the temperature control policy did enhance the performance of the CMG--at least for the first few weeks following the observance of initial indications of failure.

Some discrepancies between Barish and BCL existed concerning the level of the critical radial load. To reconcile these discrepancies a meeting was held on May 7, 1974. As a result of the meeting, Barish issued an addendum⁽¹¹⁾ to his initial report which implied reasonable agreement with the BCL predictions.

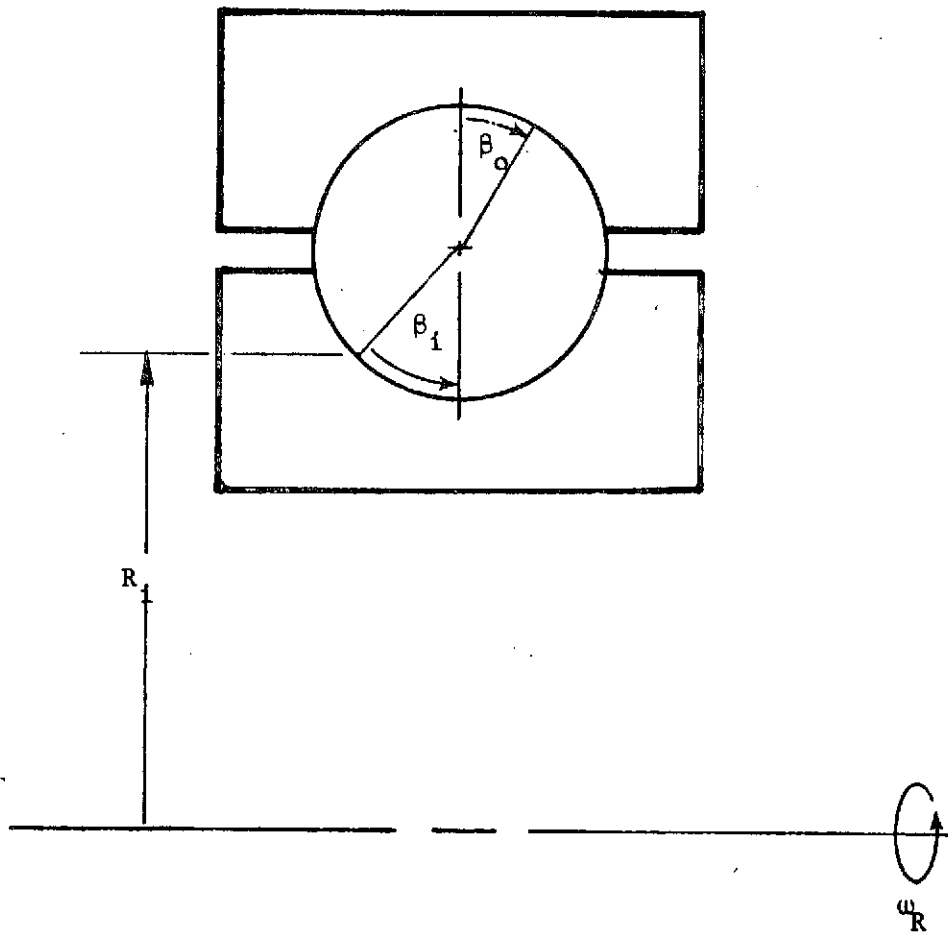


FIGURE 15. CONTACT ANGLES IN A BALL BEARING

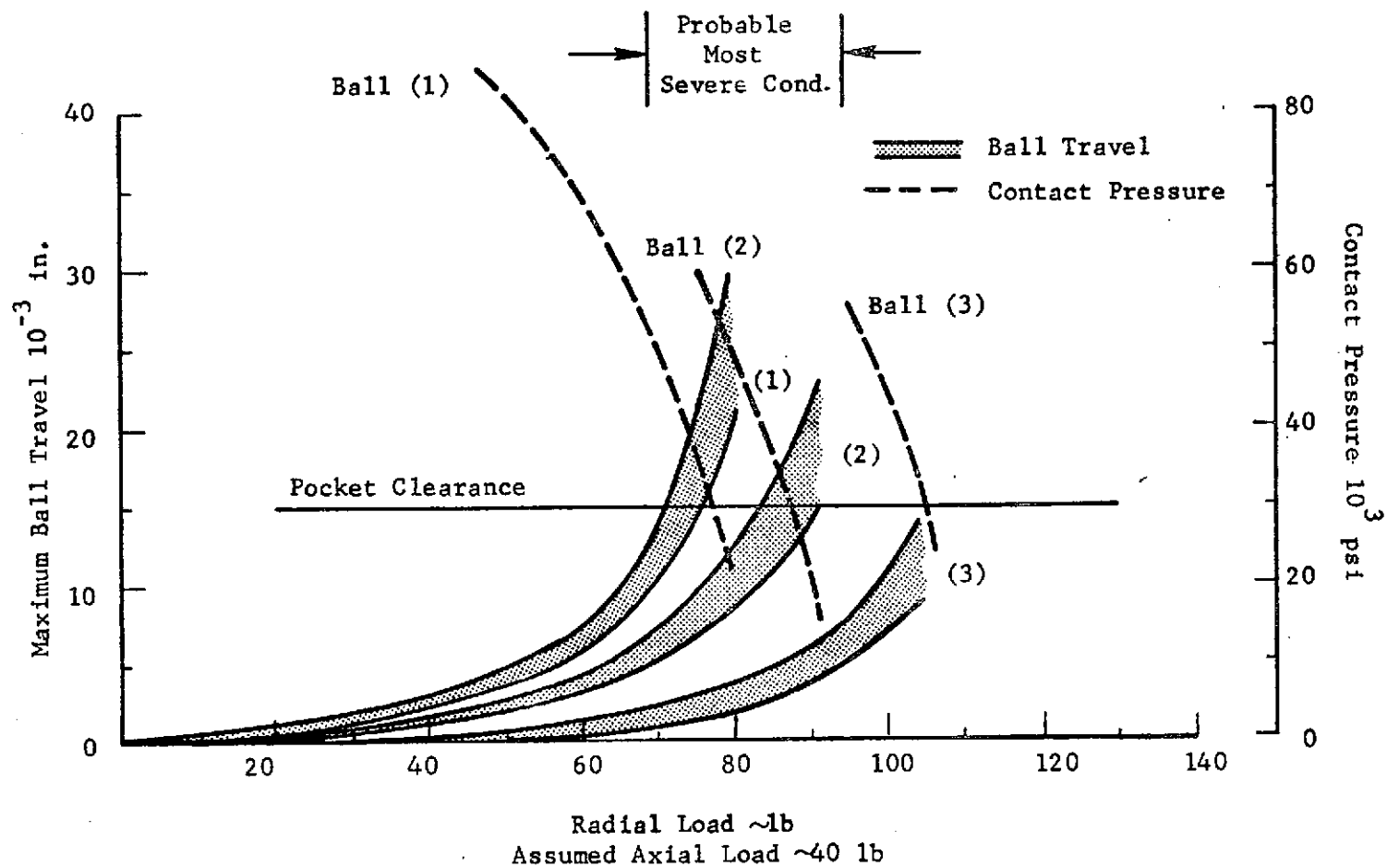


FIGURE 16. RELATIONSHIP BETWEEN BALL TRAVEL, CONTACT PRESSURE AND RADIAL LOAD FOR CMG BEARING

Inner-Race Versus Outer-Race Control

A belief which is common among bearing technologists is that one race controls the ball spin. This hypothesis is an outgrowth of the assumption of a constant coefficient of friction at the inner and outer races of contact. That is, the inner and outer contact ellipise (Figure 17) are somewhat different as a result of different ball-race conformities--both axially as well as circumferentially. With the coefficient of friction assumption, the spin torque generated on one race would be different than the torque on the other and would produce a nonequilibrium situation. Hence the race with the larger spin torque is classically assumed to "control" the spin.

In lubricated contact, the ball-race tractions are slip dependent by virtue of the basic laws of lubrication. As a result, if a ball spins more than the race, a torque producing counter rotation is generated to impede the spin. Thus, in lubricated contact, an equilibrium condition can be achieved which allows for spinning relative to both races and, therefore, race control is not operative. Nevertheless, under starved-lubrication conditions, race control can still occur and this will affect the operation of the bearing and the level of BSV.

The maximum BSV occurs just prior to radial unloading of the balls (which occurs at around 70 lb)--regardless of the race-control situation. Therefore, Figure 17 is, at least, qualitatively valid despite the particular prejudices one may have toward the race-control concept.

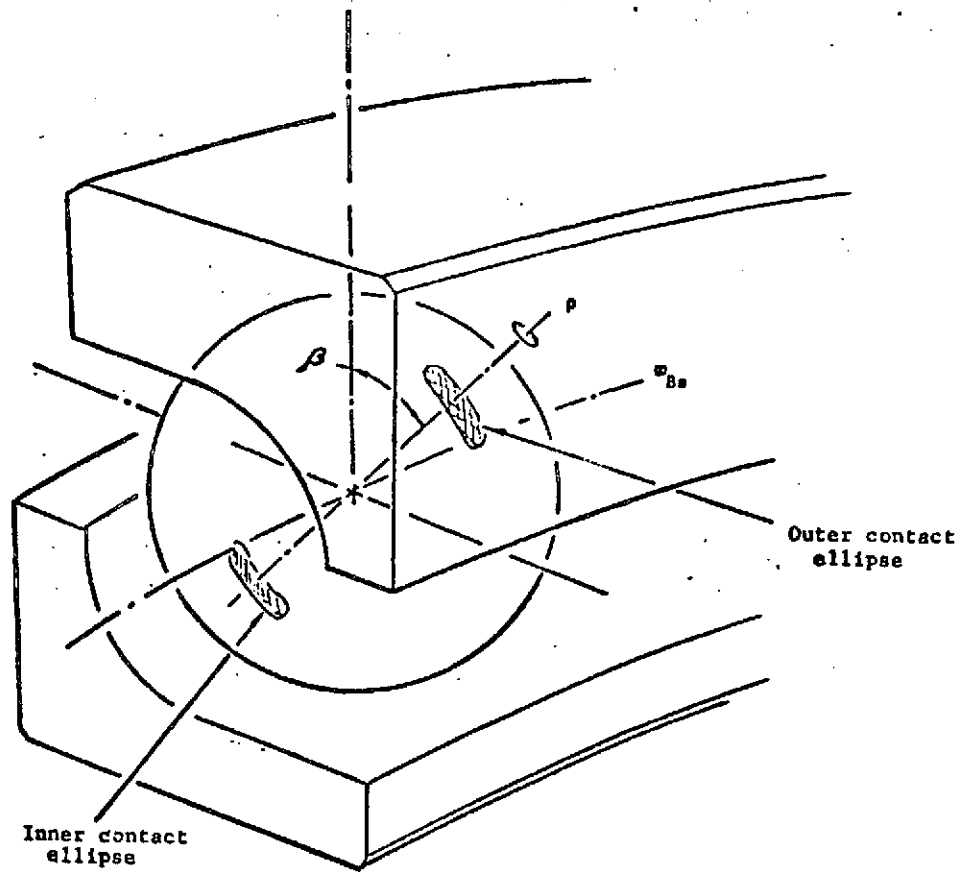


FIGURE 17. INNER AND OUTER RACE CONTACT ELLIPSES

TASK IV. RETAINER STABILITY ANALYSIS

The principal purpose of this task was to appraise, analytically, the likelihood that retainer instabilities may arise in the bearing which would contribute to the problem of torque glitches. To accomplish this task use was made of (1) a bearing dynamics computer program to compute bearing motions, and (2) a stability criterion for retainer-ball interactions.

Description of BASDAP and Input Data

For the investigation of possible retainer instabilities, the motions of the various elements of the gyroscope bearing were analyzed using a bearing dynamics computer program named BASDAP. BASDAP couples elasticity, rigid body dynamics, and hydrodynamic calculations for the balls, races, and the retainer in the bearing.

BASDAP computes ball-, retainer pocket-, and race-contact conditions as a function of time, operating speed, temperature, load, bearing design parameters, dimensional tolerances, and lubricant properties. The retainer is considered as a 6-degree-of-freedom rigid rotator with two principal axes of inertia in a plane perpendicular to the axis of the cylinder forming the inner surface of the retainer. Computer outputs include positions of the balls and retainer, lubricant-film thickness, ball-race, ball-pocket, and retainer-race tractions, stresses, and total bearing torque.

The input data required for the computer program which describe the bearing and lubricant parameters for the Skylab CMG bearing are shown in Table 1.

In the computer program, stepwise (in time) calculations are made to determine the motions and positions of the bearing elements. The convergence or divergence of the numerical computations is subject to an appropriate choice of time step which in turn depends on the magnitude of the forces acting within the bearing. For this case, the tractive forces at the ball-race contacts can be very large because of the high viscosity of the lubricant ($\sim 10^6$ cp) in the contact region. It was found that very small time steps of the magnitude of 3×10^{-5} secs were required to produce satisfactory numerical results.

TABLE 1. INPUT DATA FOR COMPUTER EVALUATIONS OF BEARING PERFORMANCE

	Numerical Value
Number of balls	15
Ball diameter, in.	0.3125
Ball mass, lb	0.00448
Pitch radius, in.	0.92
Race curvatures, percent	
Outer	0.53
Inner	0.516
Speed (inner race), rpm	9000
Cage mass, lb	0.0088
Cage width, in.	0.423
Cage land width, in.	0.05
Pocket clearance (diametral), in.	0.0165
Cage/inner race clearance (diametral), in.	0.006
Contact angle, degrees	15
Preload, lb	40
Lubricant viscosity (for KG-80), cp	250
Viscosity at ball-race contact conditions, cp	10^6
Pressure coefficient of viscosity, psi^{-1}	0.77×10^{-5}

To assess the possibility of unstable cage operation, suitable plots of cage motion with time have been made. These are described as follows.

- (1) The axial position of the outside edge of the retainer, XSV, as viewed from a spatially fixed point.
- (2) The radial position of the outside edge of the retainer, ZSV, as viewed from a spatially fixed point.
- (3) The in-plane center-of-mass position of the retainer, i.e., ZS versus YS. The point (ZS= 0, YS = 0) corresponds to the geometric center of the bearing.

These plots are shown and discussed in the following section.

Evaluations of Retainer Motions

In order to perform an evaluation of the retainer motions three principal cases were run. These are characterized as follows:

- Case 1. Establishment of a baseline condition for bearing operation using the inputs described in Table 1.
- Case 2. Assessment of the effect of temperature on the bearing operation by reducing by an order of magnitude the ball-race contact-zone viscosity.
- Case 3. Assessment of the effect on retainer motions of poor lubrication conditions at the ball-retainer and race-retainer contacts.

For all of the cases considered, the ball-race lubricant film thickness was nominally 45 μ in. which is taken to be representative of the lubrication condition in the actual bearing.

For the conditions of Case 1, the nominal bearing torque was computed to be 1.5 in.-oz. Graphs of the resulting retainer motions for Case 1 are shown in Figures 18 through 20. The quantities XSV, ZSV, and ZS and YS which describe the retainer motions have been defined above. The total time spanned is 1.86 msec which is equivalent to 43 degrees rotation of the ball group. It is noted that, with respect to the fixed spatial position, there is a low-amplitude wobble in the retainer and its frequency is approximately 410 Hz. The computed data did not indicate any abnormally large changes in torque level as a result of the retainer wobble.

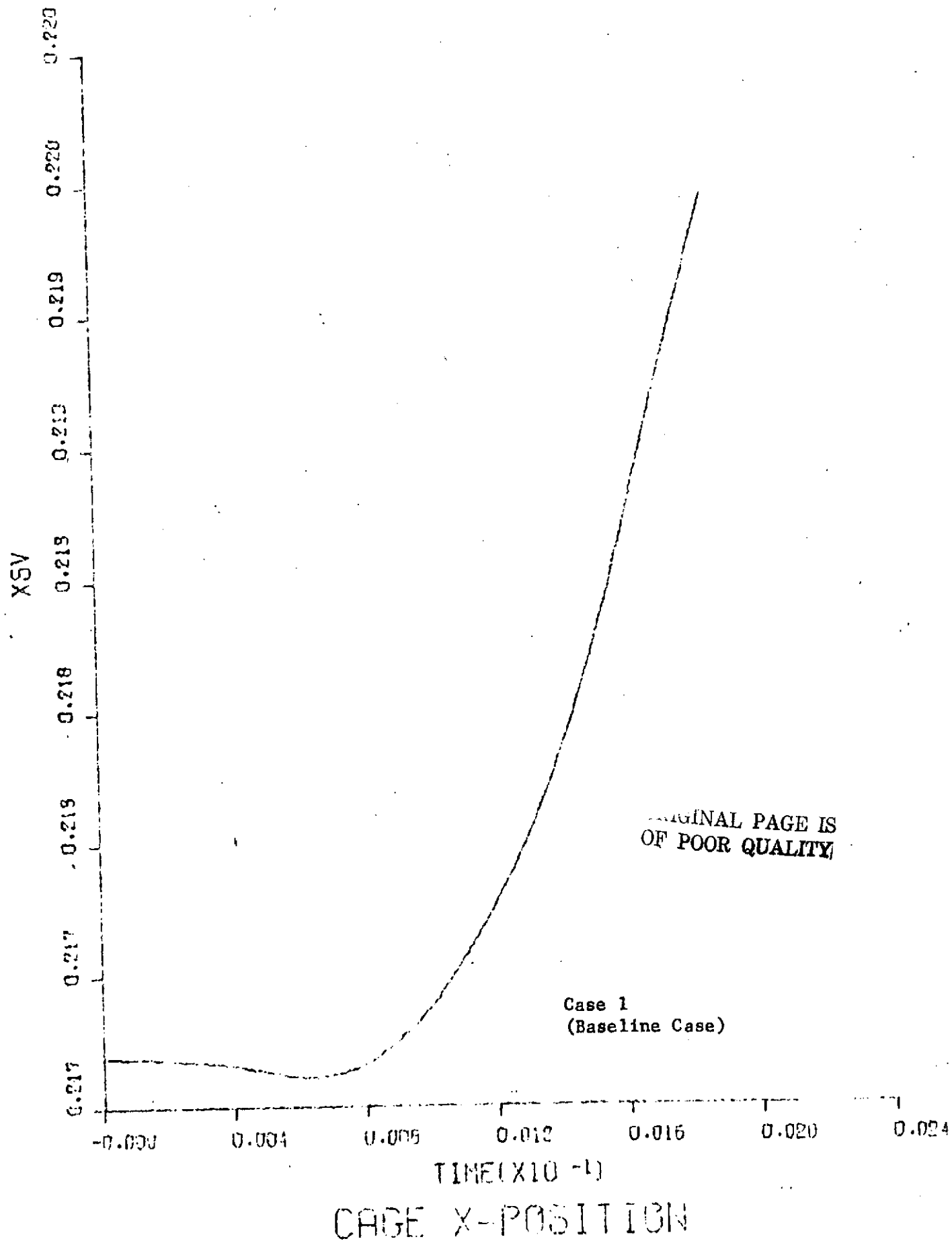


FIGURE 18. PREDICTED AXIAL POSITION OF RETAINER EDGE VERSUS TIME

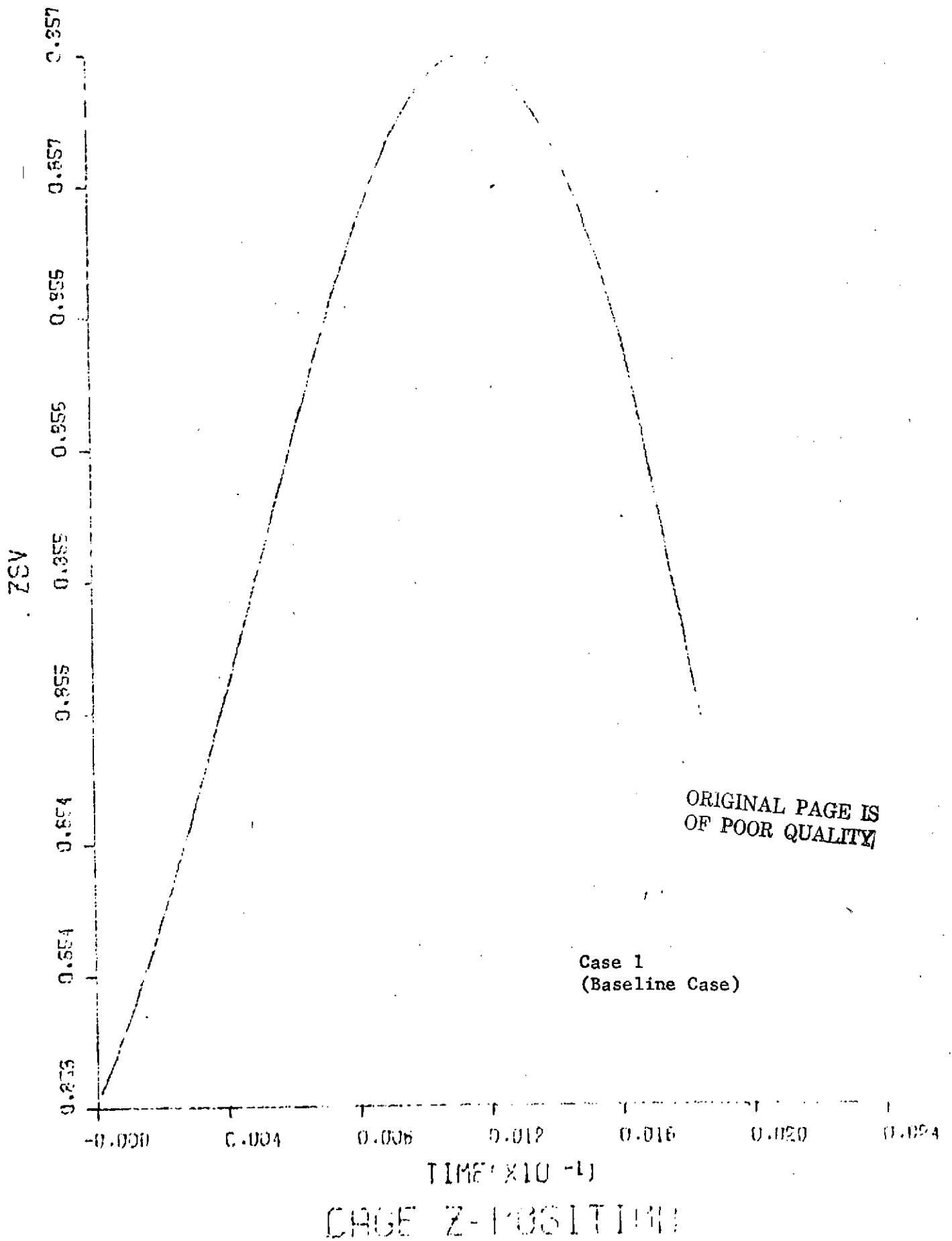
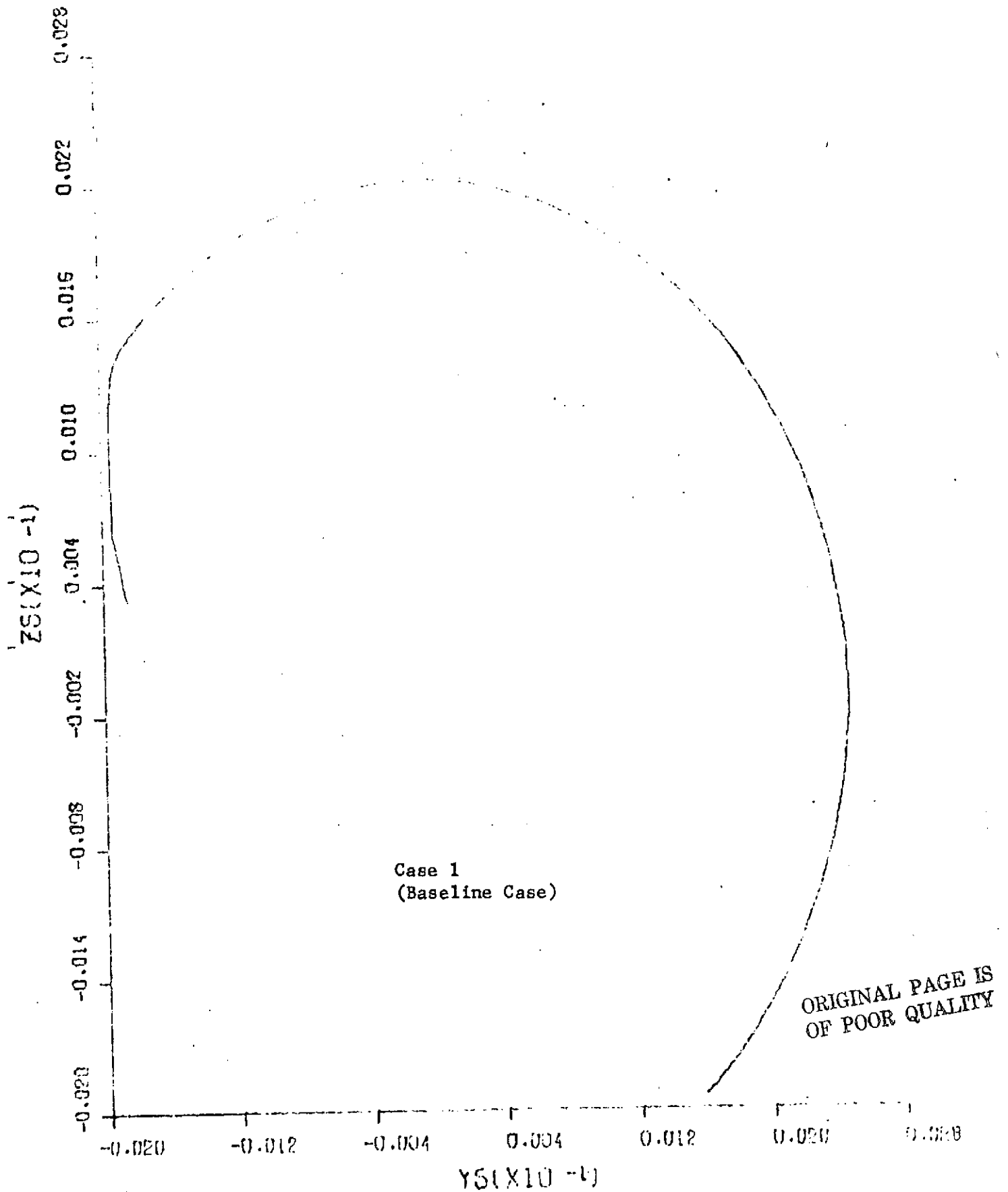


FIGURE 19. PREDICTED RADIAL POSITION OF RETAINER EDGE VERSUS TIME



IN-PLANE ORBIT MOTION

FIGURE 20. IN-PLANE ORBIT OF RETAINER CENTER OF MASS

In Case 2, the effect of temperature on bearing operation was assessed by assuming that the contact-zone viscosity of the lubricant in the ball/race contacts was lower by at least an order of magnitude than that in the previous case (contact zone viscosity $\approx 6 \times 10^4$ cp). As a result of the increase in temperature (lowering of the contact-zone viscosity), the bearing torque is lower than that for Case 1 and is nominally 0.5 in.-oz. The retainer position plots comparable to those shown for Case 1 are given in Figures 21 through 23. The plots for Cases 1 and 2 are quite similar, the main difference being that in Case 2 the amplitude of the wobble is less than that in Case 1. The wobble frequency remains essentially the same for both cases. The computer data showed also that the torque tended to be more uniform over time for the higher temperature condition than for the previous case.

For Case 3, it was assumed that conditions of very poor lubrication existed at the ball-retainer and retainer-race contacts. For this condition, the computer output data indicated that the balls were intermittently impacting the retainer. The retainer position curves for this case are shown in Figures 24 through 26. Of interest is the ZSV-versus-time plot (in-plane radial motion). The overall frequency of the wobble is approximately 1200 Hz. The small-scale step changes in the retainer position are attributed to the ball-retainer impacts. The significant factor determined in this case is that large variations in bearing torque are noted in the computer data (ranging in the order of 9 in.-oz). From this it is concluded that poor lubrication at the various retainer contact locations can result in an indication of a large change in torque time in the bearing over a relatively long period of time.

In general, the results obtained from the three principal cases run do not show any propensity for the retainer to operate in an unstable mode--even when an initial condition was assumed in which the rotational speed of the retainer was greater (by 10 percent) than that of the ball group velocity. The retainer quickly reached a velocity close to that of the ball group and wobble amplitudes were generally quite low. In a separate study being performed for the U.S. Government, a stability criterion which establishes the conditions under which the retainer may become unstable is being investigated. The criterion is based on the notion of the

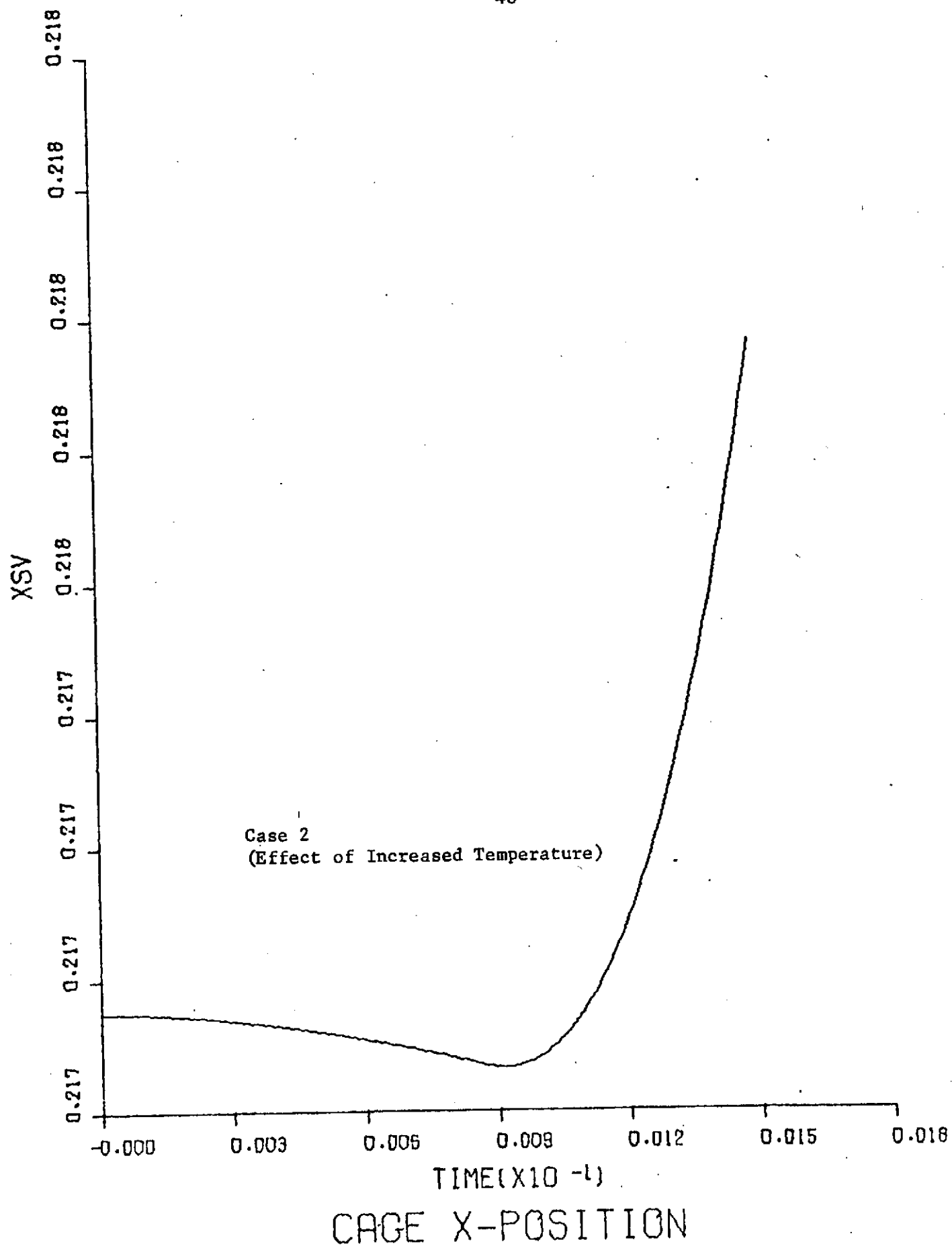


FIGURE 21. PREDICTED AXIAL POSITION OF RETAINER EDGE VERSUS TIME

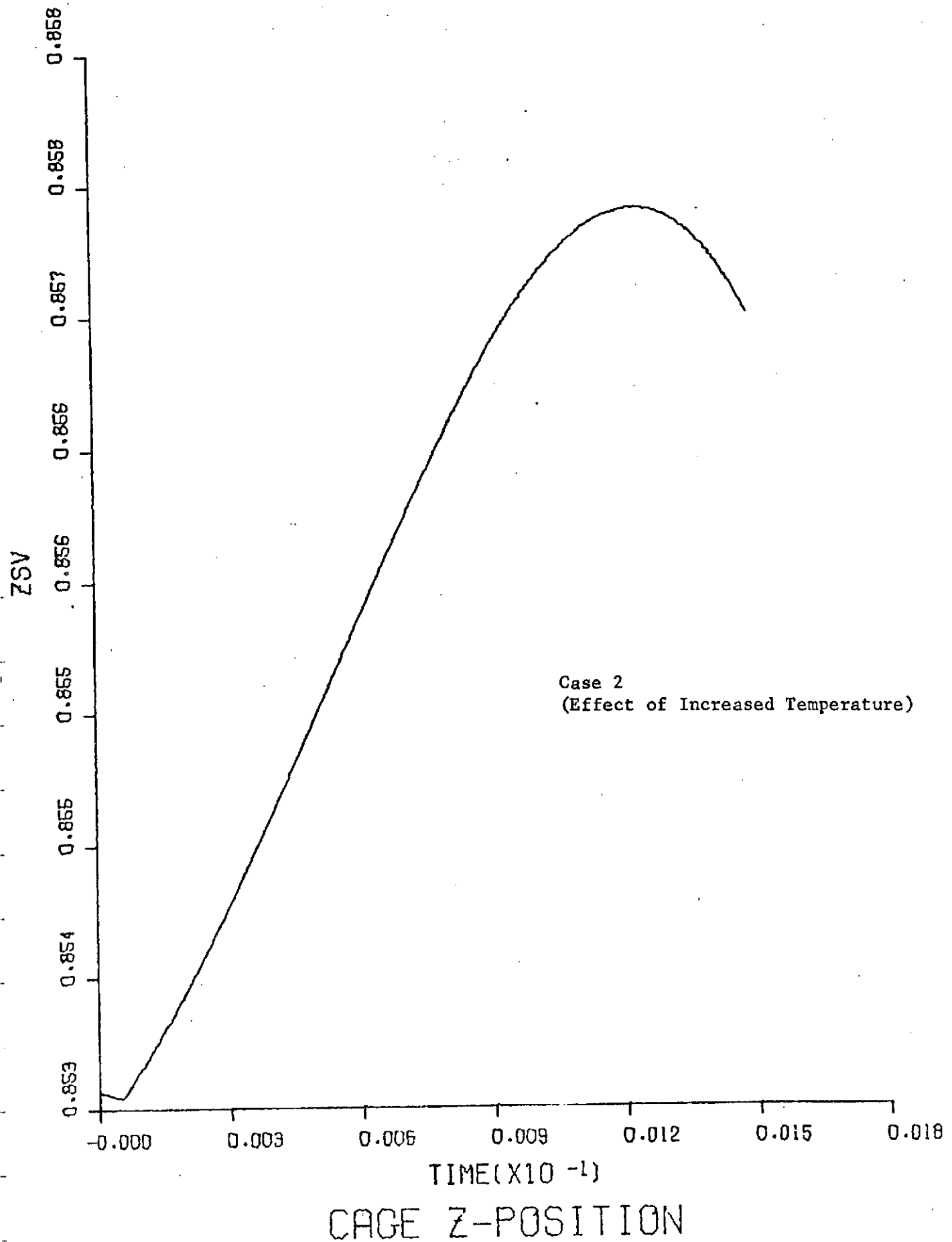
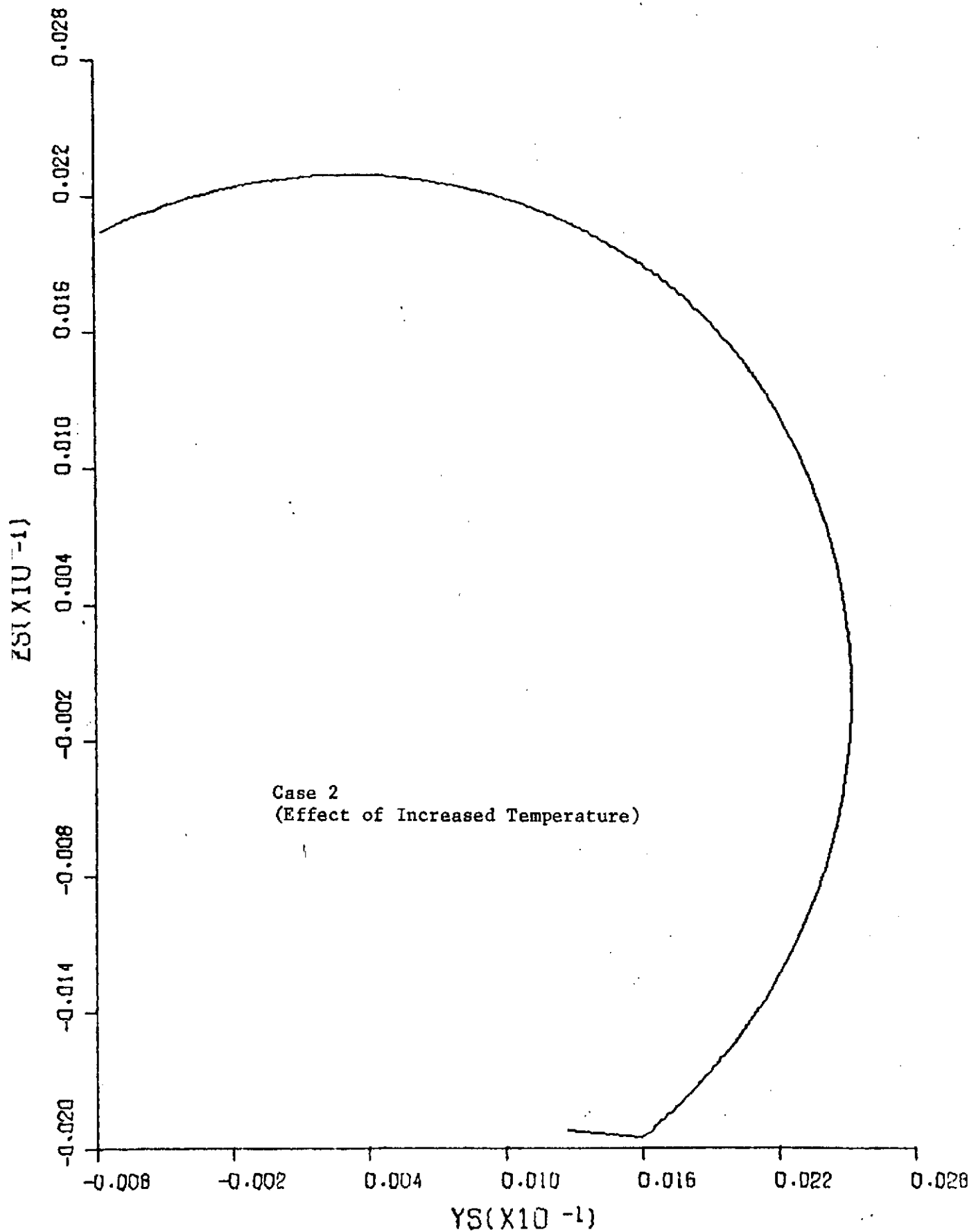


FIGURE 22. PREDICTED RADIAL POSITION OF RETAINER EDGE VERSUS TIME



IN-PLANE CM MOTION

FIGURE 23. IN-PLANE ORBIT OF RETAINER CENTER OF MASS

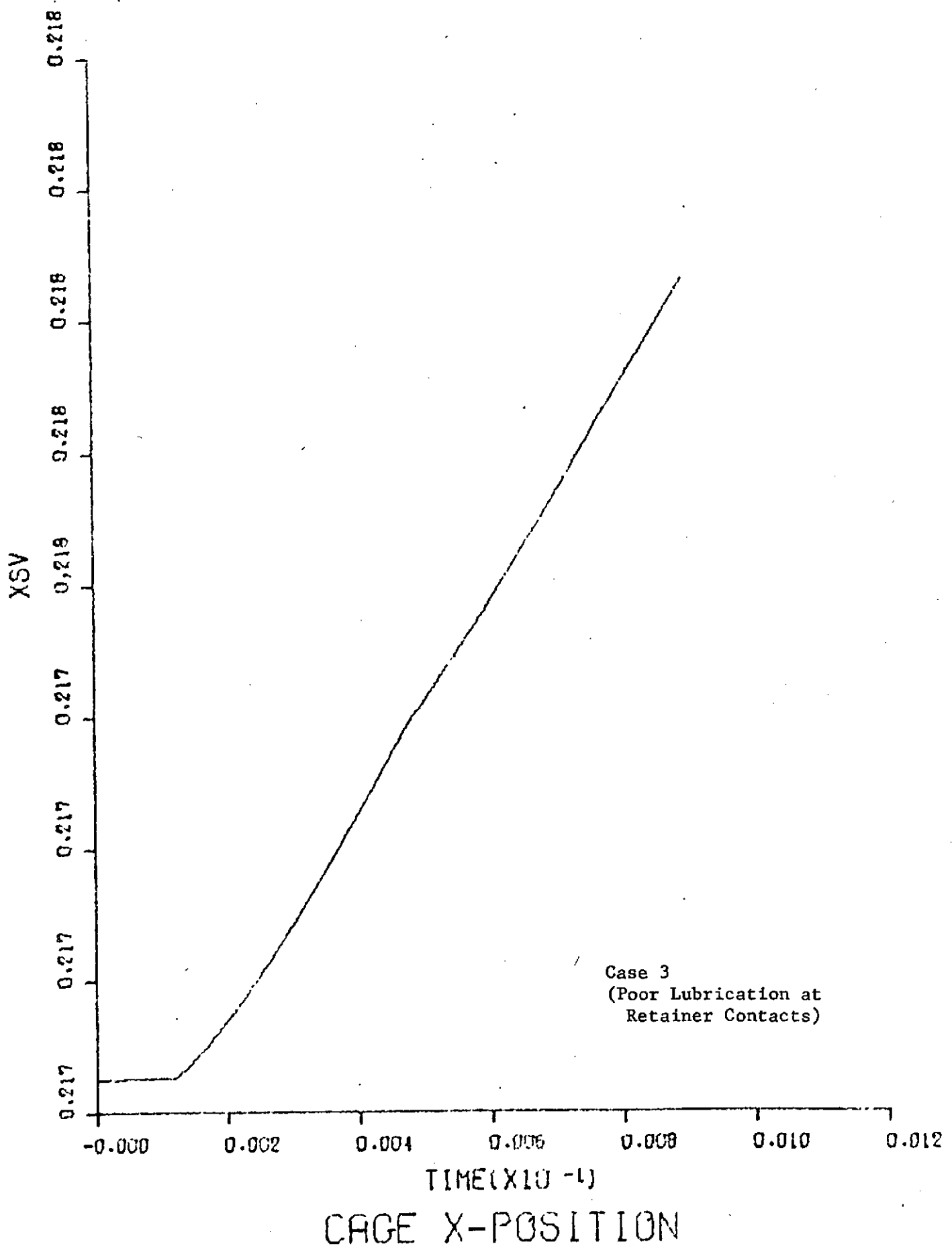


FIGURE 24. PREDICTED AXIAL POSITION OF RETAINER EDGE VERSUS TIME

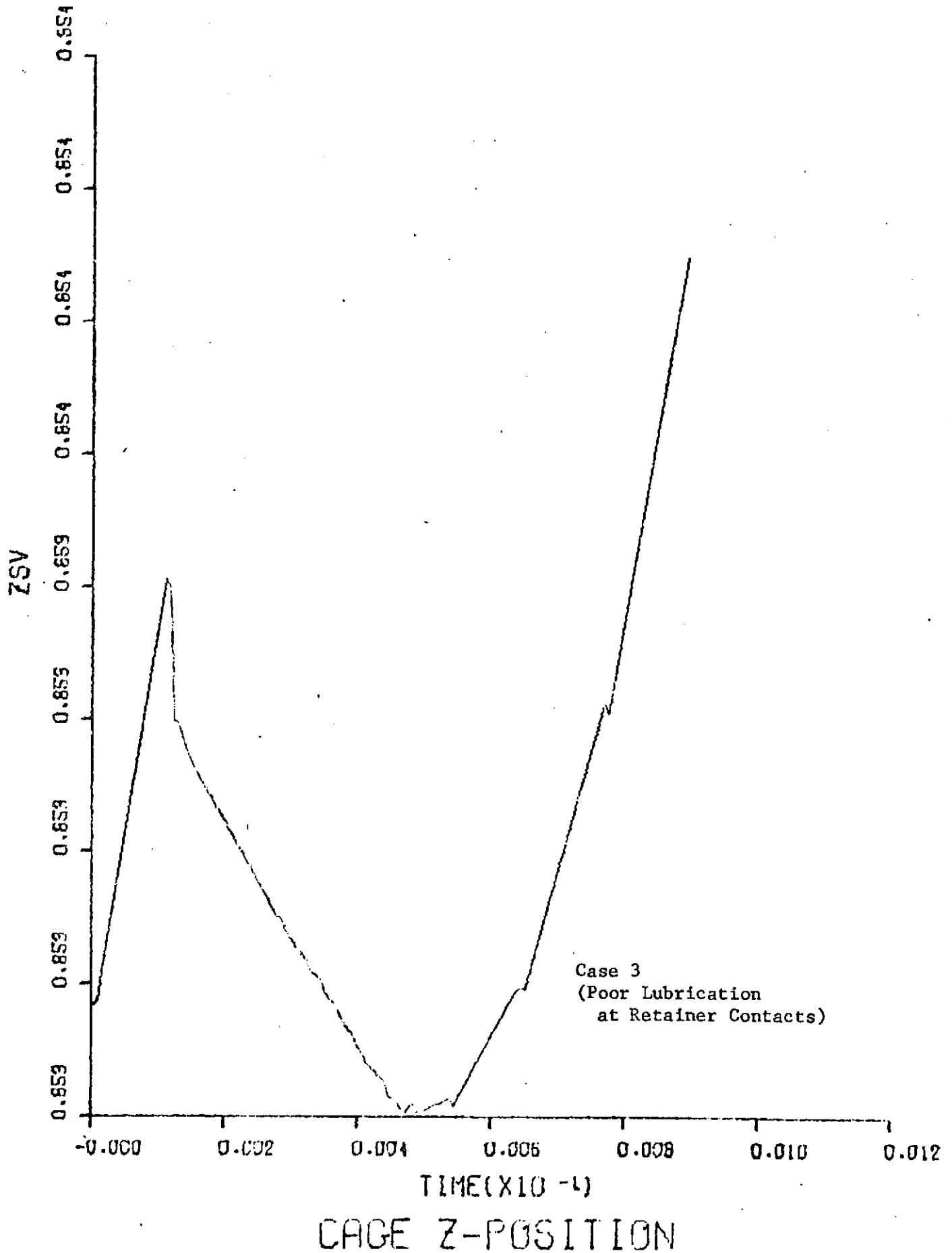


FIGURE 25. PREDICTED RADIAL POSITION OF RETAINER EDGE VERSUS TIME

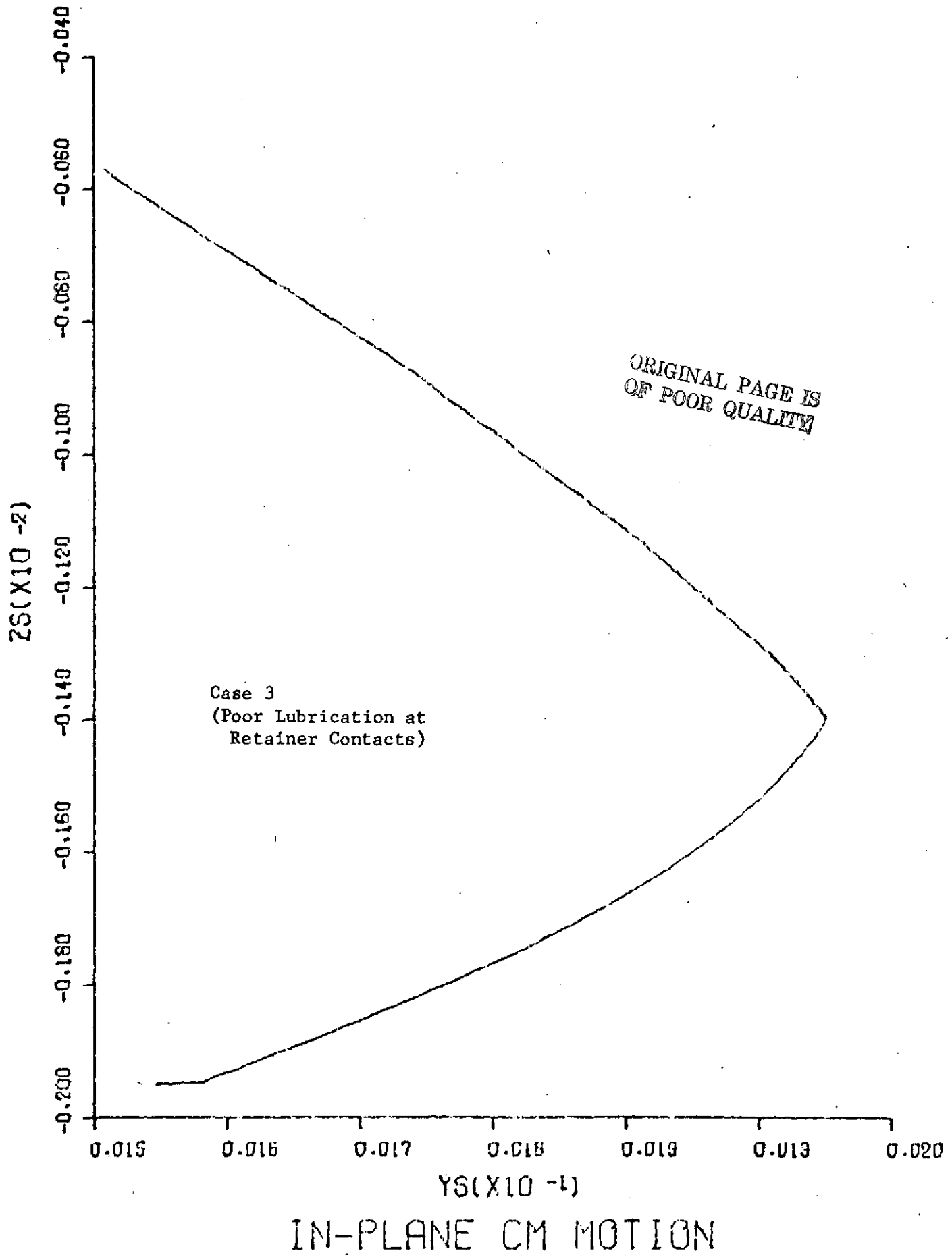


FIGURE 26. IN-PLANE ORBIT OF RETAINER CENTER OF MASS

capability of the retainer to damp ball-retainer impacts. The stability criterion takes the form

$$\frac{4C_{\mu}^2}{M_c C_s} > 1 \quad ,$$

where C_{μ} is a measure of the viscous damping at the ball-race contact (dependent on lubricant viscosity and ball-race film thickness and contact pressure), C_s is a measure of the spring rate of the retainer and ball, and M_c is the mass of the retainer. By using this criterion, it is computed that the film thickness between the ball and race must be of the order of 7 μ in. or less. Such a condition could be obtained under very starved lubrication conditions and may cause retainer instabilities. These, in turn, may give rise to large torque variations as noted for the case of poor lubrication at the ball-retainer and race-retainer contacts.

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Columbus Laboratories
505 King Avenue
Columbus, Ohio 43201
Telephone (614) 299-3151
Telex 24-5454

October 7, 1974

Mr. Richard Abramowitz
Navigation and Controls Division
Bendix Corporation
Teterboro, New Jersey 07608

Dear Rick:

In connection with the Skylab CMG investigation, the question of gravitational effects on oil feed and distribution was raised. We have studied this problem from the standpoint of the performance of the oil feed/distribution process, and offer the following comments for your consideration.

The dominant forces in the oiler itself are centrifugal and are of such a magnitude that the presence of gravity is probably insignificant. That is, it is unlikely that oiler feed rates, measured at the oiler lip, will be influenced by gravity. After leaving the oiler lip, a drop of oil is slung across a short gap and is caught under a lip on the retainer. Since the gap is short compared to the kinetic energy of the droplet, this step in the feed process is probably not appreciably different in zero-g. It is conceivable, however, that an aerosol could be formed, either by slinging very small droplets off the lip or by larger drops breaking up upon impact with the retainer and not sticking. I would expect the critical drop size for aerosol formation to be somewhat larger in zero-g than in one-g. This, however, appears to be a second order effect and is likely of little consequence.

Once on the retainer surface, a moiety of oil is conducted to the critical friction interfaces by a combination of centrifugal and surface forces. None of these transport mechanisms is strongly affected by gravity, unless oil builds up in sufficient quantity (or thickness) to begin to flow under the influence of gravity. Such a critical flow point might be just adjacent to the ball track, where the ball-pass squeeze effect, acting in concert with wear debris dams and nonwetting due to transfer films from the phenolic retainer, brings about a quantity of oil that can develop a sufficient meniscus to respond to a gravitational force. Such menisci have been observed in our laboratory experiments. If this situation develops (most probably on the outer race where centrifugal forces are absent), gravity could have an effect on the quantity of oil in the ball track.

For axis-vertical operation in a one-g field, the bottom oiler is feeding upward against the gravitational field. The slump effect, adjacent to the ball track, would tend to deplete the lubricant supply to the outer race of the bottom bearing. Furthermore, the oil feed on the bottom bearing is dependent upon surface transport from the shoulder to the groove. Very thin

October 7, 1974

films will creep under the influence of surface forces and, thus, thin film transport will tend to be independent of gravity. For thick films having a sufficient meniscus, gravity will tend to retard feed to the lower bearing and enhance feed to the top, in an axis-vertical configuration. The position of the retainer (thought to be critical with regard to the alignment of the oil transport holes relative to the outer-race groove) will be influenced by gravity. In an axis-vertical, one-g test, the retainer of the lower bearing will tend to drop down into its extreme outboard position and the oil holes will probably be located over the shoulder. The primary oil feed to the outer-race ball groove will then be by means of the surface transport and flow, viz., the mechanisms just discussed.

Of the two possible areas of gravitational influences considered, aerosol formation is likely to be most detrimental in zero-g. Flow effects, on the other hand, will tend to degrade the oil-supply process in one-g under certain conditions. Of the two processes, I think the flow effects is most important and, therefore, the one-g tests are conservative relative to orbital performance. Aerosol formation, and oiler droplet size is an interesting area for potential study in the refinement of the oiler concept for long missions. However, I believe this was a relatively unimportant influence on the Skylab CMG performance.

If you have any further questions, please feel free to call me on Extension 2942.

Very truly yours,



David K. Snediker
Senior Project Manager

DKS:lk

To: Engineering File - MT-15,785
From: T. Wheelock

Issue: Original
Date: January 6, 1975

LITERATURE SEARCH
SKYLAB CMG BEARING INVESTIGATION

Prepared by: *T. F. Wheelock*

T. Wheelock

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY

1.0 REQUIREMENT FOR FURTHER INVESTIGATION OF BEARING OPERATION IN ORBIT

The limited amount of information as to the nature of the bearing failure in CMG 1 indicated that failure modes in orbit could be produced by conditions not readily duplicated by on-earth testing.

In order to further investigate phenomena connected with orbital conditions of weightlessness and hard vacuum, a study of published material was conducted.

The NASA Scientific and Technical Information Facility, College Park, Maryland was contacted in order to obtain a bibliography of applicable documents. They were to provide a literature search concerning the topic, "Lubricants in Zero Gravity and High Vacuum" (NASA Literature Search No. 25393, Mar. 23, 1974). This literature search was reviewed and 59 documents were selected as most likely to contain information relative to the specific application. Some were duplications issued under various organizations. The documents reviewed are listed in Appendix 1. Most reports originated in the mid-1960's as this was the period of maximum activity (and minimal actual experience).

2.0 REVIEW OF DOCUMENTS

The documents contained results of investigation analysis and testing of various lubricants and apparatus. Many concerned testing and comparison of various types of oils, greases, solid lubricants, metal films, transfer films, and combinations of these in hard vacuum (10^{-5} to 10^{-10} torr). *Ref (20), (25), (27), (30), (32), (33). Some contained results

*Circled no's. refer to documents listed in Appendix I.

of life testing under various conditions simulating the space environment (12, 8, 36, 30, 34, 35). The effects of other factors of the space environment such as primary and secondary radiation (5, 33, 40), micro meteoroid impacts (25, 40) and weightlessness (3, 14, 31), were also discussed. A few document actual results of operation in orbit (17, 42). Topics of particular interest included coefficient of friction investigations (9, 17, 27, 42, 22), oxide formation, surface chemistry and adhesion (15, 16, 40, 22, 20), evaporation loss (10, 13, 33, 16, 40, 24), test procedures used (18, 33, 34, 26, 25), and labyrinth seals (20, 3, 18, 25).

The entire concept of simulation of orbital conditions for life testing on Earth was questioned in one report (18). One report described high speed 6000 RPM life testing with greases (16).

There was little information concerning the operation of large preloaded bearings with oil lubrication only at high RPM in orbital conditions (which is the case for the ATM CMG). What data was applicable did not cover areas actually evaluated with operating CMG's in testing which simulated space conditions. An evaluation of the ATM CMG bearings and lubricants was made by Bendix Research Lab (Ref (45)).

Some papers had useful discussions and analyses of lubricant failure modes. The evaporation loss of KG-80 under various conditions of vacuum and temperature was determined from Ref. (1, 17, 19, 24). The effects of radiation were referred to, and judged not to be a significant consideration.

The effects of weightlessness ("0" gravity) were covered only briefly and affects were limited to feed from lubricant reservoirs, changes in bearing loads, convective cooling and suggestions that migration and circulation of lubricant could be affected. The theoretical effects of weightlessness on bulk fluids was thoroughly investigated in Ref.

(43). These results do not directly apply to thin films or droplets of lubricant which occur in the bearings. Ref.

(44) gave some insight into formation and adhesion of films.

3.0 CONCLUSIONS

The results of the literature search indicate that while generalized testing programs for conditions to be expected in a space application are useful, data for each application must be determined under specific conditions of the actual application. While many test programs were documented, there were none actually dealing with a comparable CMG application. These documents probably exist (the Bendix Document Ref. (45) is an example); however, they are not readily available for general information. There is surprisingly little information generally available for the behavior of lubricants (for any specific fluids) in a weightless condition. This is considered (possibly rightly so) a minor factor in the lubrication mechanisms; however, the quantitative effect on oil migration with the system remains in question. The actual behavior of the lubricant and lubrication system will only be completely defined when actual behavior in orbit is observed or bearings which have been in orbit are recovered.

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4.0 APPENDIX I - PAPERS REVIEWED

1. A67-38142 - Lubrication and Wear Fundamentals for High Vacuum Applications
2. A72-18596 - Transfer Film Formation By Lubricative Composites
3. A70-34158 - Controlled Leakage Sealing of Bearings for Fluid Lubrication in a Space Vacuum Environment
4. A70-34157 - Lubrication of DC Motors, Slip Ring Bearings and Gears for Long Life Space Applications
5. A66-25770 - Lubricants for the Space Environment
6. A69-26893 - A Flywheel for Stabilizing Space Vehicles
7. A69-18564 - A Short Survey of European Work on Lubrication in a Vacuum
8. A69-18562 - Lubrication of Bearings and Gears for Operation In a Space Environment
9. A64-19124 - Lubricant Behavior in a High Vacuum
10. A74-12319 (73LC-5C-2) - Grease Lubrication of Rolling Bearings in Spacecraft
11. A73-13012 - Lubrication for Long Terms Aerospace Applications
12. A68-21757 - Lubrication in Space Vacuum (10^{-10} Torr)
13. A73-14366 - The Evaporation of Various Lubricant Fluids in Vacuum
14. A70-45862 - Design Criteria for Spacecraft Fluid Mechanical Systems

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15. A68-34169 - Some Aspects of the Surface Chemistry of Adhesion and of Friction
16. A64-24312 - Long-Duration Lubrication Studies in Simulated Space Vacuum
17. A68-30318 - Lubricants for the Space Environment
18. NASA TMX-52208 - Lubrication and Bearing Problems in Vacuum of Space
19. NASA TMX-52271 - Lubrication and Wear Fundamentals for High Vacuum Applications
20. AO635078 (Lockheed) - Lubricant Evaluation
21. AO765300 - Lubricants for Use in the Space Environment (Bibliography)
22. AO690618 - Effect of Oxide Film on the Friction of 52100 Steel in Vacuum
23. NASA TMX-63606 - Design and Development of a Radial Load Ball Bearing Test System and Some Preliminary Results
24. Lubrication and Bearing Problems in the Vacuum of Space
25. N64-19952 - Lubrication in Space
26. N66-15241 - In Flight Test To Determine Space Environmental Effects on Friction Wear, and Lubrication of Materials
27. AO240580 - AEOC-TOR-63-154 - Lubrication Requirements for Space Environments
28. FL 210/48 - A439678 - Aerospace Lubrication Research Survey

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29. 65LC-19 - Sliding Behavior of Some Lattice Layer Compounds in Ultra-high Vacuum
30. 1810-TN-7 (A.I.L.) - Problems of Lubrication in Space
31. 66AM 7A2 - Status of Lubricants for Manned Spacecrafts
32. 66AM3C2 - Advanced Aerospace Greases
33. 62LC-4 - Lubricant Evaluation for Bearing Systems Operating in Special Environments
34. 66AM-7A3 - Lubricant Life Tests on Ball Bearings for Space Applications
35. Report from Lab. Eng. Feb. '61 - Lubrication in Space
36. 69 LUB-30 - Vacuum Evaluation of Lubricants and Techniques for Space Exposed Components
37. A70-34145 - Unique Mechanism Features of ATS Stabilization Bearing Package
38. N64-29508 NTIS - MRC for JPL - Survey of Frictional Problems in Spacecraft
39. AD403441 - Cold Welding in a Vacuum on Annotated Bibliography
40. N62-12809 (JPL) - Behavior of Materials In Space Environments
41. AD744903 (Lockheed 1972) - Low Torque Control Bearings
42. AIAA - A632049-7 - Results of Ranger 1 Flight Friction Experiment

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43. Weightless; Physical Phenomena and Biological Effects.
Symposium on Physical and Biological Phenomena Under
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Edition Western Per.
44. NASA TMX-64835 - Skylab III and IV Science Demonstrations,
Preliminary report Bannister, Tommy C. NASA George
C. Marshall Space Flight Center, Alabama, Mar. 1974.
45. Bearing Verification Tests for the Bendix Apollo
Telescope Mount, 3 April 1968, Bendix Research
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Issue: Original
Date: Jan. 6, 1975

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Page 8

5.0

APPENDIX II

NASA Literature Search Number

25393
Part II (Limited Distribution References)

LUBRICANTS IN ZERO GRAVITY AND HIGH VACUUM

March 29, 1974

SCOPE: References pertinent to the above subject.

PERIOD: 1962 to date shown above

FORMAT: Citations arranged by Accession Number

NUMBER OF
CITATIONS: Machine Search - 21

This Literature Search was prepared in response to an individual's specific request, and contains references selected to meet the requester's needs.

NASA SCIENTIFIC AND TECHNICAL INFORMATION FACILITY

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73X70614 NOT AVAILABLE

72X82668* NASA-CR-129366 QPR-5 NAS8-26619 MRI PRJ. 3470-E
72/08/15 42 PAGES UNCLASSIFIED DOCUMENT NASA
RESEARCH ON BEARING LUBRICANTS FOR USE IN HIGH VACUUM QUARTERLY
PROGRESS REPORT, 1 MAY - 31 JUL. 1972
A/CAMPBELL, M. E.; B/LAVIK, M. T.
MIDWEST RESEARCH INST., KANSAS CITY, MO. (ENGINEERING SCIENCES
DIV.)

/ *HIGH VACUUM/ *JOURNAL BEARINGS/ *LUBRICANTS/ GRAPHITE/ MOLYBDENUM
SULFIDES/ SPACE SHUTTLES/ THERMOGRAVIMETRY/ THIN FILMS

72X82554* NASA-CR-129455 NAS3-6479 66/12/29 71 PAGES
UNCLASSIFIED DOCUMENT NASA
HYDRODYNAMIC JOURNAL BEARING PROGRAM MONTHLY PROGRESS REPORT, 29
NOV. - 29 DEC. 1966
A/NICHOLS, H. E.; B/RICHARDS, W. D. C.; C/ROSSEACH, R. J.;
D/COSTALES, D. J.
GENERAL ELECTRIC CO., CINCINNATI, OHIO. (MISSILE AND SPACE DIV.)
/ *JOURNAL BEARINGS/ *LUBRICANTS/ *WEIGHTLESSNESS/ PARAMETERIZATION/
PERFORMANCE TESTS/ ROTATING SHAFTS/ VIBRATION TESTS

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UNCLASSIFIED DOCUMENT NASA
RESEARCH ON BEARING LUBRICANTS FOR USE IN HIGH VACUUM QUARTERLY
PROGRESS REPORT, 1 MAY - 31 JUL. 1971
A/CAMPBELL, M. E.; B/LAVIK, M. T.
MIDWEST RESEARCH INST., KANSAS CITY, MO. (ENGINEERING SCIENCES
DIV.)
/ *HIGH TEMPERATURE LUBRICANTS/ *HIGH VACUUM/ *LUBRICANT TESTS/
BINDERS (MATERIALS)/ SOLID LUBRICANTS

66X83462* NASA-CR-77162 NAS3-2534 65/00/00 30 PAGES
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MATERIALS FOR POTASSIUM LUBRICATED JOURNAL BEARINGS STATUS REPORT,
APR. 22 - MAY 22, 1965
GENERAL ELECTRIC CO., CINCINNATI, OHIO. (SPACE POWER AND
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<1965< 30 P REFS
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JOURNAL/ LUBRICATION/ POTASSIUM/ PRESSURE/ VIBRATION/ WEAR

71X10339*# ISSUE 2 PAGE 68 CATEGORY 15 NASA-CR-103060
NAS8-25318 71/01/00 101 PAGES UNCLASSIFIED DOCUMENT GOVT.+
CONTR.

DEVELOPMENT AND CHARACTERIZATION OF LUBRICANTS FOR USE NEAR NUCLEAR
REACTORS IN SPACE VEHICLES

(DEVELOPMENT OF RADIATION STABLE WIDE TEMPERATURE RANGE LUBRICANTS
FOR USE NEAR NUCLEAR REACTOR IN SPACECRAFT ENVIRONMENT)

A/AKAWIE, R. I.; B/BRITTAIN, D.; C/GARDCS, M. N.; D/JONES, J. R.
HUGHES AIRCRAFT CO., CULVER CITY, CALIF. (AEROSPACE GROUP.)

/*LUBRICANTS/*NUCLEAR REACTORS/*RADIATION TOLERANCE/*SPACECRAFT
ENVIRONMENTS/ ADDITIVES/ AROMATIC COMPOUNDS/ CHEMICAL REACTIONS/
PRODUCT DEVELOPMENT/ SILOXANES/ SPACECRAFT LUBRICATION

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DDC-TAS-70-8-2-VOL-2 70/03/00 209 PAGES

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2)

DEFENSE DOCUMENTATION CENTER, ALEXANDRIA, VA.

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TEMPERATURE RESEARCH/ LOW PRESSURE/ RADIATION EFFECTS/ TEMPERATURE
EFFECTS/ ULTRAHIGH VACUUM/ WEIGHTLESSNESS

69X10065*# ISSUE 1 PAGE 57 CATEGORY 15 NASA-CR-98069
NAS8-21165 68/07/22 55 PAGES UNCLASSIFIED DOCUMENT GOVT.+ CONTR.
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SUMMARY REPORT, 23 APR. 1967 - 22 JUL. 1968

(SOLID LUBRICANT FILMS FOR BALL BEARINGS AND SPUR GEARS USED IN HIGH
VACUUM ENVIRONMENTS)

A/CAMPBELL, M. E.; B/GADDIS, D. H.; C/HASS, H.

MIDWEST RESEARCH INST., KANSAS CITY, MO.

/*BALL BEARINGS/*GEARS/*HIGH VACUUM/*SOLID LUBRICANTS/ AEROSPACE
ENVIRONMENTS/ LUBRICATION/ POLYIMIDES

68X18983# ISSUE 20 PAGE 1765 CATEGORY 15 AFML-TR-68-76
AD-832431 AF 33/615/-3781 68/04/00 75 PAGES UNCLASSIFIED
DOCUMENT GOVT.+ CONTR.

FRICTION AND WEAR OF SOLID MATERIALS SLIDING IN ULTRAHIGH VACUUM AND
CONTROLLED, GASEOUS ENVIRONMENTS SUMMARY REPORT, 1 MAR. 1967 - 29 FEB.
1968

(APPARATUS IMPROVEMENT FOR TESTING OF FRICTION AND WEAR OF SOLID
MATERIALS SLIDING IN ULTRAHIGH VACUUM AND CONTROLLED GASEOUS
ENVIRONMENTS)

A/HALTNER, A. J.

GENERAL ELECTRIC CO., PHILADELPHIA, PA. (MISSILE AND SPACE DIV.)

WRIGHT-PATTERSON AFB, OHIO AF MATER. LAB.

/*SLIDING FRICTION/*SOLIDS/*ULTRAHIGH VACUUM/*VACUUM

APPARATUS/*WEAR TESTS/ ACETONE/ CONTROLLED ATMOSPHERES/ CRYOGENICS/ GAS
DYNAMICS/ LUBRICATION/ MERCURY (METAL)/ OUTGASSING/ SPACE ENVIRONMENT
SIMULATION/ TORQUE

68X16149# ISSUE 13 PAGE 1150 CATEGORY 15 R68SD6 AD-829345
68/03/00 50 PAGES UNCLASSIFIED DOCUMENT GOVT.+ CONTR.

LUBRICANTS FOR THE SPACE ENVIRONMENT

(LUBRICANTS FOR SPACE ENVIRONMENT APPLICATIONS)

A/FLOM, D. G.; B/HALTNER, A. J.

GENERAL ELECTRIC CO., PHILADELPHIA, PA. (MISSILE AND SPACE DIV.)

PRESENTED AT THE MATER. CONF., PHILADELPHIA, 31 MAR. - 4 APR.

1968

/*LUBRICANTS/*SPACECRAFT LUBRICATION/ FRICTION REDUCTION/
LUBRICATION SYSTEMS/ SELF LUBRICATING MATERIALS

68X11301# ISSUE 4 PAGE 311 CATEGORY 15 AFFDL-TR-66-181
AD-822649 67/09/00 34 PAGES UNCLASSIFIED DOCUMENT GOVT.+ CONTR.

INVESTIGATION OF HIGH TEMPERATURE BEARING MATERIALS UNDER
ATMOSPHERIC AND SIMULATED SPACE CONDITIONS TECHNICAL REPORT, MAY 1965
- JUN. 1966

(SLIDING FRICTION AND WEAR TESTS OF CARBURIZED AND NITRIDED CHROME
COMPOSITES AND ALUMINA AT TEMPERATURES TO 2000 DEG F IN SIMULATED SPACE
CONDITIONS)

A/KRYSIAK, J. E.

AIR FORCE SYSTEMS COMMAND, WRIGHT-PATTERSON AFB, OHIO. (AF FLIGHT
DYNAMICS LAB.)

/*ALUMINUM OXIDES/*CHROMIUM ALLOYS/*COMPOSITE MATERIALS/*SLIDING
FRICTION/*SPACECRAFT LUBRICATION/*WEAR TESTS/ HIGH TEMPERATURE TESTS/
REFRACTORY MATERIALS/ SPACE ENVIRONMENT SIMULATION

67X18631# ISSUE 17 PAGE 1686 CATEGORY 15 AFFDL-TR-66-134
AD-813599 AF 33/615/-2549 67/01/00 93 PAGES UNCLASSIFIED
DOCUMENT GOVT.+ CONTR.

AEROSPACE BEARING - ANALYTICAL INVESTIGATION
(BALL BEARING AND SEPARATOR DESIGN FOR LIGHTLY LOADED HIGH SPEED
BEARINGS LUBRICATED WITH DRY FILM FOR SPACE APPLICATION)
A/GIANNOTTI, R. J.; B/LEVEILLE, A. R.; C/PELLETIER, R. R.
GENERAL MOTORS CORP., BRISTOL, CONN. (NEW DEPARTURE-HYATT BEARINGS
DIV.)

WRIGHT-PATTERSON AFB, OHIO, AF FLIGHT DYN. LAB., JAN. 1967 93 P
REFS

/*BALL BEARING/*MATERIAL TESTING/*SEPARATOR/*SPACE ENVIRONMENTAL
LUBRICATION/ BALL/ BEARING/ COMPUTER/ DESIGN/ DRY/ ENVIRONMENT/ FILM/
FRICTION/ GRAPHITE/ LOADING/ LUBRICATION/ MATERIAL/ MATHEMATICS/ MODEL/
MOLYBDENUM/ ROLLING/ SILICATE/ SIMULATION/ SLIDING/ SODIUM/ SPACE/
SULFIDE/ TESTING/ THRUST/ VACUUM

66X18789# ISSUE 15 PAGE 1631 CATEGORY 15 ER-6627-3

IR-8-304/III/ AD-481062 AF 33/615/-2749

66/04/15 37 PAGES UNCLASSIFIED DOCUMENT GOVT.+ CONTR.
HIGH TEMPERATURE EXTRUSION LUBRICANTS INTERIM ENGINEERING PROGRESS
REPORT, 1 JAN. - 31 MAR. 1966

(HIGH TEMPERATURE EXTRUSION LUBRICANTS FOR AEROSPACE MATERIAL
APPLICATIONS)

A/HAVERSTRAW, R. C.

TRW, INC., CLEVELAND, OHIO. (MATERIALS AND PROCESSES DEPT.)

WRIGHT-PATTERSON AFB, OHIO, AF MATER. LAB., 15 APR. 1966 37 P
REFS

/*EXTRUSION/*HIGH TEMPERATURE LUBRICANT/*SPACE ENVIRONMENTAL
LUBRICATION/ ALUMINUM/ CALCIUM/ COMPOUND/ ENVIRONMENT/ GRAPHITE/ HEAT/
HIGH TEMPERATURE/ LUBRICANT/ MAGNESIUM/ MATERIAL/ REACTIVITY/
RESISTANCE/ SODIUM/ SPACE/ TESTING/ TUNGSTEN/ ZIRCONIUM

66X12667 ISSUE 5 PAGE 508 CATEGORY 15 64/09/30 20 PAGES
UNCLASSIFIED DOCUMENT GOVT.+ CONTR.

LONG-DURATION LUBRICATION STUDIES IN SIMULATED SPACE VACUUM

(LUBRICATION OF SLIDING AND ROLLING ELEMENT BEARINGS IN SIMULATED
SPACE VEHICLES)

A/BROWN, R. D.; B/BURTON, R. A.; C/KU, P. M.

SOUTHWEST RESEARCH INST., SAN ANTONIO, TEX.

IN AFSC PROC. OF THE USAF-SWRI AEROSPACE BEARING CONF. 30 SEP.
1964 P 132-153 REFS /SEE X66-12657 05-15/

/*BALL BEARING/*ROLLING CONTACT BEARING/*SPACE ENVIRONMENTAL
LUBRICATION/*SPACE SIMULATION/*SPACE VEHICLE/ BALL/ BEARING/ CONTACT/
ENVIRONMENT/ LUBRICATION/ ROLLING/ SIMULATION/ SLIDING/ SPACE/
TEMPERATURE/ TEST/ VACUUM/ VEHICLE

66X12662 ISSUE 5 PAGE 507 CATEGORY 15 64/09/30 20 PAGES
UNCLASSIFIED DOCUMENT GOVT.+ CONTR.

STUDIES OF LUBRICATING MATERIALS IN VACUUM
(LUBRICATING MATERIALS IN VACUUM ENVIRONMENT, AND FRICTION AND WEAR
STUDIES)

A/BUCKLEY, D. H.; B/JOHNSON, R. L.; C/SHIKERT, M. A.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO.

IN AFSC PROC. OF THE USAF-SWRI AEROSPACE BEARING CONF. 30 SEP.
1964 P 60-79 REFS /SEE X66-12657 05-15/

/*FRICTION/*SPACE ENVIRONMENT/*SPACE ENVIRONMENTAL
LUBRICATION/*WEAR/ ALLOY/ BINARY/ CONTACT/ ENVIRONMENT/ EVAPORATION/
FILM/ LUBRICATION/ MATERIAL/ POLYMER/ SLIDING/ SOLID/ SPACE/
VACUUM

66X12661 ISSUE 5 PAGE 507 CATEGORY 15 64/09/30 13 PAGES
UNCLASSIFIED DOCUMENT GOVT.+ CONTR.

AIR FORCE MATERIALS LABORATORY SOLID FILM LUBRICATION RESEARCH
(BEARING LUBRICATION REQUIREMENTS OF AEROSPACE VEHICLES, AND SOLID
FILM LUBRICANTS FOR EXTREME ENVIRONMENTAL CONDITIONS)

A/MC CONNELL, B. D.

AIR FORCE SYSTEMS COMMAND, WRIGHT-PATTERSON AFB, OHIO. (AIR FORCE
MATERIALS LAB.)

IN ITS PROC. OF THE USAF-SWRI AEROSPACE BEARING CONF. 30 SEP.
1964 P 47-59 REFS /SEE X66-12657 05-15/

/*AEROSPACE VEHICLE/*BEARING/*FILM/*SOLID LUBRICANT/*SPACE
ENVIRONMENTAL LUBRICATION/ AEROSPACE/ ENVIRONMENT/ FRICTION/ HEAT/
LUBRICANT/ LUBRICATION/ METAL/ REQUIREMENT/ SOLID/ SPACE/ VEHICLE/
WEAR

66X12660 ISSUE 5 PAGE 513 CATEGORY 17 64/09/30 18 PAGES
UNCLASSIFIED DOCUMENT GOVT.+ CONTR.

THE ROLE OF ATMOSPHERIC COMPONENTS IN FRICTION, WEAR, AND BOUNDARY
LUBRICATION

(ROLE OF ATMOSPHERIC COMPONENTS IN FRICTION AND WEAR OF METALS AND
ALLOYS, AND BOUNDARY LUBRICATION - REACTIONS IN SPACE ENVIRONMENT)

A/CAMPBELL, W. E.

RENSSELAER POLYTECHNIC INST., TROY, N.Y.

IN AFSC PROC. OF THE USAF-SWRI AEROSPACE BEARING CONF. 30 SEP.
1964 P 29-46 REFS /SEE X66-12657 05-15/

/*ALLOY/*ATMOSPHERE/*BOUNDARY LUBRICATION/*FRICTION/*METAL/*SPACE
ENVIRONMENT/*WEAR/ ADSORPTION/ BOUNDARY/ COMBUSTION/ COMPONENT/
ENVIRONMENT/ INTERACTION/ INTERFACE/ LUBRICATION/ OXYGEN/ PRODUCT/
REACTION/ SPACE/ WATER

65X14898 ISSUE 12 PAGE 954 CATEGORY 32 65/01/00 13 PAGES
UNCLASSIFIED DOCUMENT GOVT.+ CONTR.
SELECTION OF MATERIALS FOR OPTIMUM PERFORMANCE
(STRUCTURAL MATERIAL SELECTION FOR OPTIMUM SPACECRAFT PERFORMANCE IN
APPROPRIATE SPACE ENVIRONMENTS)
A/VATTA, T. F.
LOCKHEED MISSILES AND SPACE CO., SUNNYVALE, CALIF.
IN ITS SPACE MATER. HANDBOOK JAN. 1965 P 601-641 REFS /SEE
X65-14878 12-34/
/*SPACE ENVIRONMENT/*SPACECRAFT PERFORMANCE/*STRUCTURAL MATERIAL/
ADHESIVE/ COMPONENT/ CONTROL/ ELECTRONIC/ ENVIRONMENT/ LUBRICATION/
MATERIAL/ MISSION/ OPTICAL/ PERFORMANCE/ SEAL/ SPACE/ SPACECRAFT/
STRUCTURAL/ SYSTEM/ THERMAL

65X14889 ISSUE 12 PAGE 907 CATEGORY 15 65/01/00 87 PAGES
UNCLASSIFIED DOCUMENT GOVT.+ CONTR.
MATERIALS FOR LUBRICATED SYSTEMS
(SPACE ENVIRONMENTAL LUBRICATION MECHANISMS STUDY WITH VARIOUS
LUBRICANTS AND SELF-LUBRICATING MATERIALS)
A/CLAUSS, F. J.; B/YOUNG, W. C.
LOCKHEED MISSILES AND SPACE CO., SUNNYVALE, CALIF.
IN ITS SPACE MATER. HANDBOOK JAN. 1965 P 209-295 REFS /SEE
X65-14878 12-34/
/*LUBRICATION/*SELF-LUBRICATING MATERIAL/*SPACE ENVIRONMENTAL
LUBRICATION/ ENVIRONMENT/ GRAPHITE/ GREASE/ GROUND/ LEAD/ MATERIAL/
METAL/ MOLYBDENUM/ OIL/ PLASTICS/ SELF-ACTING/ SCFT/ SPACE/ SULFIDE/
TIME

64X15773# ISSUE 18 CATEGORY 17 FL-210-53 AD-439844 63/00/00
327 PAGES UNCLASSIFIED DOCUMENT GOVT.+ CONTR.
PROCEEDINGS OF SPACE LUBRICATION CONFERENCE
(SPACE ENVIRONMENTAL LUBRICATION - SELF-LUBRICATING MATERIAL,
FRICTION AND WEAR, ADHESION BONDING, AND GREASE LUBRICANTS -
CONFERENCE)
A/EVANS, H. E.
DEFENSE DEPT., WASHINGTON, D.C.
SEP. 1963 327 P REFS CONF. HELD AT NEW YORK, 3 MAY 1963.
CO-SPONSORED BY NASA AND AM. SOC. OF LUBR. ENGRS.
/*CONFERENCE/*SELF-LUBRICATING MATERIAL/*SOLID LUBRICANT/*SPACE
ENVIRONMENT/*SPACE ENVIRONMENTAL LUBRICATION/ ADHESION/ BONDING/
ENVIRONMENT/ FRICTION/ FUEL/ GREASE/ MATERIAL/ OXIDIZER/ SPACE/ WEAR

62X11005 ISSUE 9 CATEGORY 17 ASD-TDR-62-541 62/06/00

UNCLASSIFIED DOCUMENT MONITORED

(LUBRICATION IN SPACE ENVIRONMENTS)

A/ADAMCZACK, R. L.; B/BENZING, R. J.; C/SCHWENKER, H.

DIRECTORATE OF MATERIALS AND PROCESSES, AERONAUTICAL SYSTEMS DIV.,
WRIGHT-PATTERSON AFB, OHIO.

DIRECTORATE OF MATERIALS AND PROCESSES, AERONAUTICAL SYSTEMS
DIV., WRIGHT-PATTERSON AFB, OHIO. LUBRICATION IN SPACE ENVIRONMENTS.
<FINAL REPORT>. ROBERT L. ADAMCZACK, ROBERT J. BENZING, AND HERBERT
SCHWENKER. JUNE 1962. 18 P. /ASD-TDR-62-541/ AVAILABLE TO U.S.
GOVERNMENT AGENCIES AND CONTRACTORS ONLY.

/*LUBRICATION/*SPACE ENVIRONMENT/ ALIPHATIC/ BEARING/ COOLANT/
DEBRIS/ DESIGN/ ENERGY/ ENVIRONMENT/ FLUID/ GAS/ GREASE/ HEAT/
HYDRAULICS/ HYDROCARBON/ LIQUID/ LUBRICANT/ MEDIUM/ MINERAL/ OIL/
RELIABILITY/ SOLID/ SPACE/ TRANSFER

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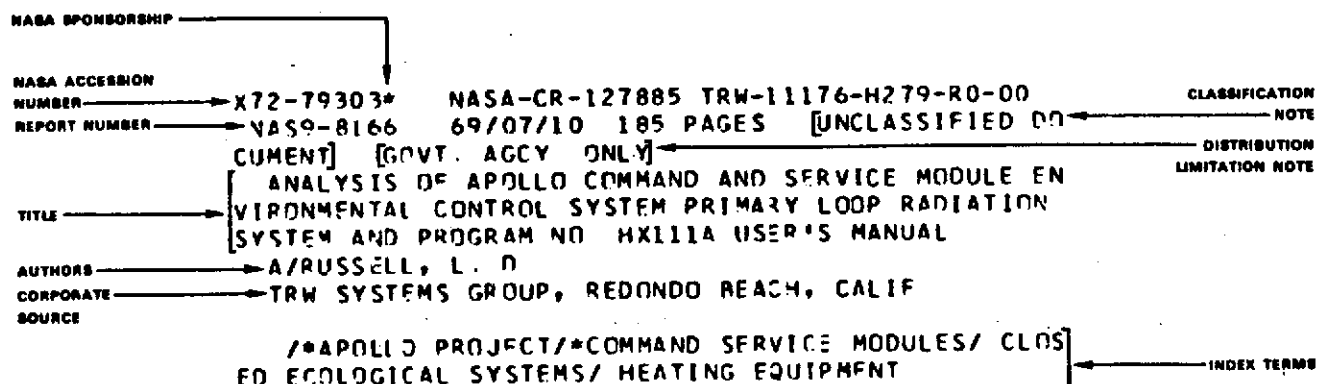


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300 000 - 361 514	X	
361 515 - 395 999		X
396 000 - 396 999	X	
397 000 - 399 999		X
AD-400 000 SERIES		
400 000 - 464 929	X	
464 930 - 489 999		X
490 000 - 492 999	X	
493 000 - 499 999		X
AD-600 000 SERIES		X
AD-800 000 SERIES		X
DDC NUMBER UNKNOWN		X
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AD-1	199 999	X	
	200 000 - 299 999	X	
AD-300 000 SERIES			
	300 000 - 361 514	X	
	361 515 - 395 999		X
	396 000 - 396 999	X	
	397 000 - 399 999		X
AD-400 000 SERIES			
	400 000 - 464 929	X	
	464 930 - 489 999		X
	490 000 - 492 999	X	
	493 000 - 499 999		X
	AD-600 000 SERIES		X
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NASA Literature Search Number

25393

LUBRICANTS IN AERO GRAVITY AND HIGH VACUUM

March 29, 1974

SCOPE: References pertinent to the above subject.

PERIOD: 1962 to date shown above

FORMAT: Citations arranged by Accession Number

NUMBER OF
CITATIONS: Machine Search - 138

This Literature Search was prepared in response to an individual's specific request, and contains references selected to meet the requester's needs.

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At the end of this search you will find a detailed explanation of the format and the data elements for the document citations. Also included is information on the availability of cited literature. Citations pertaining to security classified and/or limited distribution documents are included under a separate cover, if appropriate to your search request and user registration.

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PRINT 11/2/1-138

TERMINAL=55

72B10469* CATEGORY 4 M-FS-22119 72/08/00 UNCLASSIFIED DOCUMENT
DOMESTIC

RESEARCH ON BEARING LUBRICANTS FOR USE IN A HIGH VACUUM
(WORK, DURING THE RESEARCH PROGRAM, WAS CONCENTRATED ON LUBRICANT
DEVELOPMENT, GEAR LUBRICATION AND EVALUATION, PROVIDING COATED TEST
SPECIMENS, ADVISING NASA CONTRACTORS ABOUT SOLID LUBRICATION SPECIFIC
APPLICATIONS, AND INVESTIGATION OF NEW METHOD OF ATTACHING LUBRICATING
SOLIDS TO BEARING SURFACES BY SPUTTERING TECHNIQUE.)

A/CAMPBELL, M. E.; B/HASS, H. A/(MIDWEST RES. INST.); B/(MIDWEST
RES. INST.)

/*BEARINGS/*BINDERS (MATERIALS)/*GRAPHITE/*HIGH
VACUUM/*LUBRICANTS/*MELTING POINTS/*MOLYBDENUM DISULFIDES/*NOBLE
METALS/*PHOSPHATES/*RESEARCH AND DEVELOPMENT/*SILICATES

71B10478* CATEGORY 3 ARC-10056 71/12/00 UNCLASSIFIED DOCUMENT
DOMESTIC

IMPROVED MOLECULAR SORBENT TRAP FOR HIGH-VACUUM SYSTEMS
(CLOSED CYCLE REFRIGERATION LOOP IN WHICH TRAYS HOLDING MOLECULAR
SORBENT ARE MADE TO SERVE AS COOLING BAFFLES IMPROVES THE PERFORMANCE
OF HIGH VACUUM SYSTEMS. HIGH PERFORMANCE IS OBTAINED WITH ALMOST NO
DECREASE IN PUMPING SPEED.)

A/KNECHTEL, F. D.; B/PITTS, W. C.

/*BAFFLES/*COLD WATER/*COMPRESSING/*CONDENSERS (LIQUIFIERS)/*COOLING
SYSTEMS/*FEEDBACK CONTROL/*FREQUENCY/*HIGH VACUUM/*INDUCTORS/*LIQUID
NITROGEN/*LUBRICATING OILS/*MOLECULAR
ABSORPTION/*REFRIGERANTS/*SORBENTS/*STAINLESS
STEELS/*TRAPS/*WELDING

67B10379* CATEGORY 5 M-FS-12341 67/10/00 UNCLASSIFIED DOCUMENT
DOMESTIC

MACHINE TESTS SLOW-SPEED SLIDING FRICTION IN HIGH VACUUM
(TESTING MACHINE THAT OPERATES WITHOUT ANY LUBRICATION OF THE
MACHINE ELEMENTS WITHIN THE VACUUM CHAMBER MEASURES STATIC FRICTION AND
SLIDING FRICTION AT VERY LOW SPEEDS. MOVING PARTS ARE HELD TO A
MINIMUM TO SIMPLIFY OPERATION IN THE VACUUM CHAMBER.)

A/SKYRUS, J.; B/WILKINSON, C.

/*AMPLIFIERS/*FRICTION MEASUREMENT/*HIGH VACUUM/*KINETIC
FRICTION/*LUBRICANT TESTS/*MACHINERY/*MEASURING INSTRUMENTS/*METAL
STRIPS/*MOLYBDENUM DISULFIDES/*PINS/*PLATES (STRUCTURAL
MEMBERS)/*RECORDING INSTRUMENTS/*SHAFTS (MACHINE ELEMENTS)/*SIGNAL
ANALYZERS/*SLIDING FRICTION/*STATIC FRICTION/*STRAIN GAGES/*VACUUM
APPARATUS/*WHEATSTONE BRIDGES

66B10673* CATEGORY 5 LEWIS-359 66/12/00 UNCLASSIFIED DOCUMENT
DOMESTIC

IMPROVED ROLLING ELEMENT BEARINGS PROVIDE LOW TORQUE AND SMALL
TEMPERATURE RISE IN ULTRAHIGH VACUUM ENVIRONMENT

(ROLLING ELEMENT BEARING WITH STAINLESS STEEL RACES AND ROLLING
ELEMENTS AND A POROUS BRONZE CAGE SUCCESSFULLY OPERATES IN ULTRAHIGH
VACUUM ENVIRONMENTS AT A LOW TORQUE AND WITH SMALL TEMPERATURE RISE.
ALL COMPONENTS ARE BURNISHED IN MOLYBDENUM DISULFIDE.)

A/GLENN, D. C.

/*BALL BEARINGS/*CONTAMINANTS/*COPPER ALLCYS/*ELECTRONIC
EQUIPMENT/*HIGH VACUUM/*LUBRICANTS/*MOLYBDENUM
DISULFIDES/*POLYTETRAFLUOROETHYLENE/*STAINLESS STEELS/*TEMPERATURE
GRADIENTS/*TIN ALLOYS/*TORQUE

66B10165* CATEGORY 3 LEWIS-245 66/04/00 UNCLASSIFIED DOCUMENT
DOMESTIC

GALLIUM ALLOY FILMS INVESTIGATED FOR USE AS BOUNDARY LUBRICANTS

(GALLIUM ALLOYED WITH OTHER LOW MELTING POINT METALS HAS EXCELLENT
LUBRICANT PROPERTIES OF FLUIDITY AND LOW VAPOR PRESSURE FOR HIGH
TEMPERATURE OR VACUUM ENVIRONMENTS. THE ADDITION OF OTHER SOFT METALS
REDUCES THE CORROSIVITY AND FORMATION OF UNDESIRABLE ALLCYS NORMALLY
FOUND WITH GALLIUM.)

/*BEARING ALLOYS/*COATINGS/*CORROSION PREVENTION/*FRICTION
REDUCTION/*GALLIUM ALLCYS/*HIGH TEMPERATURE ENVIRONMENTS/*HIGH
VACUUM/*INDIUM ALLOYS/*LUBRICANTS/*SLIDING FRICTION/*STAINLESS
STEELS/*TIN ALLOYS/*VAPOR PRESSURE/*WEAR INHIBITORS/*WEAR TESTS

65B10366* CATEGORY 3 JPL-SC-075 65/12/00 UNCLASSIFIED DOCUMENT
DOMESTIC

UNIQUE GEAR DESIGN PROVIDES SELF-LUBRICATION

(COMPOSITE GEAR CONFIGURATION PROVIDES A RELIABLE AUTOMATIC MEANS
FOR REPLENISHING GEAR MECHANISM LUBRICANTS THAT DISSIPATE IN THE HARSH
ENVIRONMENT OF SPACE. THE CENTER OR HUB SECTION OF THE GEAR CONSISTS
OF A POROUS, OIL IMPREGNATED MATERIAL, AND THE OUTER OR TOOTHED SECTION
HAS RADIALY DRILLED PASSAGES TO CAUSE THE OIL TO GRADUALLY FLOW TO THE
GEAR TEETH SURFACE.)

A/WINIARSKI, F. J.

/*FLOW VELOCITY/*GEAR TEETH/*GEARS/*LUBRICATING OILS/*POROUS
MATERIALS/*SELF LUBRICATION/*SPACECRAFT LUBRICATION

63B10453* CATEGORY 3 M-FS-54 64/11/00 UNCLASSIFIED DOCUMENT
DOMESTIC

MOLYBDENUM DISULFIDE MIXTURES MAKE EFFECTIVE HIGH-VACUUM LUBRICANTS

(FIVE DIFFERENT MIXTURES OF MOLYBDENUM DISULFIDE ARE FOUND TO BE
EFFECTIVE BEARING LUBRICANTS WHEN TESTED AT VERY LOW PRESSURES AND HIGH
TEMPERATURES.)

/*BEARINGS/*ENVIRONMENTS/*HIGH VACUUM/*LUBRICANT
TESTS/*LUBRICANTS/*MOLYBDENUM DISULFIDES

63B10337* CATEGORY 3 LEWIS-12 64/05/00 UNCLASSIFIED DOCUMENT
DOMESTIC

GALLIUM USEFUL BEARING LUBRICANT IN HIGH-VACUUM ENVIRONMENT
(SOLID GALLIUM IS USED AS A LUBRICANT ON BEARINGS MADE OF COMPATIBLE
MATERIALS. SUCH LUBRICANTS PERFORM WELL IN A HIGH VACUUM AND UNDER LOW
TEMPERATURE.)

A/BUCKLEY, D. H.

/*BEARINGS/*COSTS/*GALLIUM/*HIGH VACUUM/*LOW TEMPERATURE
ENVIRONMENTS/*LUBRICANTS/*STEELS

74A12319 ISSUE 2 PAGE 218 CATEGORY 15 ASLE PREPRINT 73LC-5C-2
73/10/00 8 PAGES UNCLASSIFIED DOCUMENT

GREASE LUBRICATION OF ROLLING BEARINGS IN SPACECRAFT. I

A/MAHNCKE, H. E.; B/SCHWARTZ, A. J. B/(SKF ENGINEERING AND
RESEARCH CENTER, KING OF PRUSSIA, PA.) MEMBERS, \$1.50; NONMEMBERS,
\$2.00

AMERICAN SOCIETY OF LUBRICATION ENGINEERS AND AMERICAN SOCIETY OF
MECHANICAL ENGINEERS, JOINT LUBRICATION CONFERENCE, ATLANTA, GA., OCT.
15-18, 1973, ASLE 8 P. RESEARCH SPONSORED BY THE SPERRY RAND CORP.

/*EVAPORATION RATE/*GREASES/*LUBRICANT TESTS/*ROLLER
BEARINGS/*SPACECRAFT LUBRICATION/ ACCELERATED LIFE TESTS/ SEALS
(STOPPERS)/ TEMPERATURE EFFECTS/ VACUUM EFFECTS/ VAPOR PRESSURE

74A11446# ISSUE 1 CATEGORY 18 72/12/00 13 PAGES UNCLASSIFIED
DOCUMENT

SOLID LUBRICANTS FOR SPACE TECHNOLOGY --- HIGH TEMPERATURE AND
VACUUM ENVIRONMENTS

A/MUKHERJEE, M. K. A/(INDIAN SPACE RESEARCH ORGANIZATION, VIKRAM
SARABHAI SPACE CENTRE, TRIVANDRUM, INDIA) \$1.00

WORLD CONFERENCE IN INDUSTRIAL TRIBOLOGY, 1ST, INDIAN INSTITUTE OF
TECHNOLOGY, NEW DELHI, INDIA, DEC. 11-18, 1972, PAPER. 13 P.

/*HIGH TEMPERATURE LUBRICANTS/*LUBRICANT TESTS/*SOLID
LUBRICANTS/*SPACECRAFT LUBRICATION/*VACUUM EFFECTS/ BEARINGS/ COMPOSITE
MATERIALS/ TECHNOLOGY ASSESSMENT/ THERMAL VACUUM TESTS

73A36362# ISSUE 18 PAGE 2370 CATEGORY 33 AIAA PAPER 73-746

73/07/00 13 PAGES UNCLASSIFIED DOCUMENT

MODELING THE EFFECT OF AIR AND OIL UPON THE THERMAL RESISTANCE OF A SPHERE-FLAT CONTACT.

A/YOVANOVICH, M. M.; B/KITSCHA, W. W. A/(WATERLOO, UNIVERSITY, WATERLOO, ONTARIO, CANADA); B/(WATERLOO, UNIVERSITY, WATERLOO; ATOMIC ENERGY OF CANADA, LTD., SHERIDAN PARK, ONTARIO, CANADA) MEMBERS, \$1.50; NONMEMBERS, \$2.00

AMERICAN INSTITUTE OF AERONAUTICS AND ASTRONAUTICS, THERMOPHYSICS CONFERENCE, 8TH, PALM SPRINGS, CALIF., JULY 16-18, 1973, 13 P. RESEARCH SUPPORTED BY THE NATIONAL RESEARCH COUNCIL OF CANADA.

/*AIR/*CONTACT RESISTANCE/*LUBRICATING OILS/*SURFACE PROPERTIES/*THERMAL RESISTANCE/ BEARING/ GAS-SOLID INTERFACES/ LIQUID-SOLID INTERFACES/ SPACECRAFT INSTRUMENTS/ SPACECRAFT LUBRICATION/ THERMAL VACUUM TESTS

73A24250# ISSUE 10 PAGE 1239 CATEGORY 18 73/00/00 4 PAGES IN RUSSIAN UNCLASSIFIED DOCUMENT

USAGE OF THE VNII NP-300A LUBRICANT IN GAS-LIQUID CHROMATOGRAPHY AND HIGH-VACUUM TECHNOLOGY

A/NAZAROVA, L. I.; B/NIKONOROV, E. M.; C/LULOVA, N. I.; D/VOLOBUEV, N. K.; E/KUZ'MINA, A. V. E/(VSESOUZNYI NAUCHNO-ISSLEDOVATEL'SKII INSTITUT PO PERERABOTKE NEFTI I GAZA I POLUCHENIIU ISSKUSTVENNOGO ZHIDKOGO TOPLIVA, USSR)

KHIMIYA I TEKHOLOGIYA TOPLIV I MASEL, VOL. 18, NO. 2, 1972, P. 53-56. IN RUSSIAN.

/*GAS CHROMATOGRAPHY/*HIGH TEMPERATURE LUBRICANTS/*HIGH VACUUM/*HYDROCARBONS/*LUBRICATING OILS/ GAS-LIQUID INTERACTIONS/ LUBRICANT TESTS/ THERMAL STABILITY/ VACUUM APPARATUS

73A14997 ISSUE 4 PAGE 454 CATEGORY 15 72/11/00 8 PAGES UNCLASSIFIED DOCUMENT

FRICTIONAL BEHAVIOUR OF MOLYBDENUM DISULPHIDE IN HIGH VACUUM.

A/MATSUNAGA, M.; B/HOSHIMOTO, K.; C/UCHIYAMA, Y. A/(TOKYO, UNIVERSITY, TOKYO, JAPAN); B/(NATIONAL RESEARCH INSTITUTE FOR METALS, TOKYO, JAPAN); C/(KANAZAWA UNIVERSITY, KANAZAWA, JAPAN)

WEAR, VOL. 22, NOV. 1972, P. 185-192.

/*COEFFICIENT OF FRICTION/*CONTACT RESISTANCE/*DISULFIDES/*LUBRICANT TESTS/*MOLYBDENUM SULFIDES/*SOLID LUBRICANTS/ ADHESION/ ADSORPTION/ HIGH VACUUM/ METAL BONDING/ PROTECTIVE COATINGS/ SLIDING FRICTION/ TEST EQUIPMENT/ TRANSIENT RESPONSE/ VACUUM EFFECTS/ WEAR

73A14366* ISSUE 3 PAGE 335 CATEGORY 18 ASLE PREPRINT 72LC-6C-2
NAS7-100 72/10/00 6 PAGES UNCLASSIFIED DOCUMENT

THE EVAPORATION OF VARIOUS LUBRICANT FLUIDS IN VACUUM.

A/HAMILTON, D. B.; B/OGDEN, J. S. B/(BATTELLE COLUMBUS
LABORATORIES, COLUMBUS, OHIO) MEMBERS, \$1.50; NONMEMBERS, \$2.00
AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND AMERICAN SOCIETY OF
LUBRICATION ENGINEERS, INTERNATIONAL LUBRICATION CONFERENCE, NEW YORK,
N.Y., OCT. 9-12, 1972, ASLE 6 P.

/*EVAPORATION/*GREASES/*LUBRICANT TESTS/*PERFLUORO
COMPOUNDS/*SPACECRAFT LUBRICATION/*VACUUM EFFECTS/ ETHERS/ MATERIALS
TESTS/ OUTGASSING/ TEMPERATURE EFFECTS/ VISCOSITY

73A13012 ISSUE 3 PAGE 33C CATEGORY 18 72/0C/CO 8 PAGES
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LUBRICATION FOR LONG-TERM AEROSPACE APPLICATIONS.

(SPACECRAFT LUBRICATION SYSTEMS DESIGN FOR LONG MISSION LIFETIMES,
DISCUSSING MECHANICAL DESIGN PARAMETERS AND LIQUID AND SOLID LUBRICANT
CHARACTERISTICS)

A/HINRICKS, J. T.; B/LORAN, T. J.; C/FRIEBEL, V. R. C/(BALL
BROTHERS RESEARCH CORP., BOULDER, COLO.)

IN NON-METALLIC MATERIALS SELECTION, PROCESSING AND ENVIRONMENTAL
BEHAVIOR; PROCEEDINGS OF THE FOURTH NATIONAL TECHNICAL CONFERENCE AND
EXHIBITION, PALO ALTO, CALIF., OCTOBER 17-19, 1972. (A73-13001 03-18)
AZUSA, CALIF., SOCIETY OF AEROSPACE MATERIAL AND PROCESS ENGINEERS,
1972, P. 113-120.

/*LONG TERM EFFECTS/*LUBRICATION SYSTEMS/*SERVICE LIFE/*SPACECRAFT
LUBRICATION/ AEROSPACE ENVIRONMENTS/ LUBRICATING OILS/ SOLID
LUBRICANTS/ SOLID-SOLID INTERFACES/ SPACECRAFT ENVIRONMENTS/ SYSTEMS
ANALYSIS

73A11886 ISSUE 2 PAGE 211 CATEGORY 30 72/00/00 316 PAGES
UNCLASSIFIED DOCUMENT

SPACE RESEARCH PHYSICAL AND TECHNICAL PRINCIPLES.

(RUSSIAN BOOK ON PHYSICOTECHNOLOGICAL BASIS OF SPACE RESEARCH
COVERING NEAR EARTH AND INTERPLANETARY ENVIRONMENTAL FACTORS AND
EFFECTS ON SPACECRAFT DESIGNS AND MATERIALS)

A/KROSHKIN, M. G. \$12

(TRANSLATION OF FIZIKO-TEKHNICHESKIE OSNOVY KOSMICHESKIKH
ISSLEDOVANIY, MOSCOW, IZDATEL'STVO MASHINOSTROENIY, 1969.) JERUSALEM,
ISRAEL PROGRAM FOR SCIENTIFIC TRANSLATIONS, LTD., 1972. 316 P.

/*AEROSPACE ENVIRONMENTS/*AEROSPACE SCIENCES/*EARTH
ATMOSPHERE/*INTERPLANETARY SPACE/*SPACECRAFT DESIGN/ AEROSPACE
INDUSTRY/ GEOMAGNETISM/ RADIATION DAMAGE/ SPACECRAFT COMPONENTS/
SPACECRAFT LUBRICATION/ VACUUM EFFECTS

72A38391 ISSUE 19 PAGE 2835 CATEGORY 23 72/C8/00 7 PAGES
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SELECTING MATERIALS AND HARDWARE.
(HIGH AND ULTRAHIGH VACUUM EQUIPMENT AND COMPONENTS SELECTION,
DISCUSSING GAS-SURFACE INTERACTIONS, CONTAMINATION AND CLEANING
PROBLEMS)

A/WHEELER, W. R. A/(VARIAN ASSOCIATES, PALO ALTO, CALIF.)

PHYSICS TODAY, VOL. 25, AUG. 1972, P. 52-58.

/*CLEANING/*DECONTAMINATION/*HIGH VACUUM/*SURFACE
REACTIONS/*ULTRAHIGH VACUUM/*VACUUM SYSTEMS/ ENVIRONMENTAL CONTROL/
GASKETS/ JOINTS (JUNCTIONS)/ LUBRICANTS/ OUTGASSING/ VACUUM PUMPS/
VALVES

72A18596 ISSUE 6 PAGE 824 CATEGORY 15 71/00/00 13 PAGES
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TRANSFER FILM FORMATION BY LUBRICATIVE COMPOSITES.

(SELF LUBRICATING POLYTETRAFLUOROETHYLENE AND POLYIMIDE COMPOSITES
TRANSFER FILM FORMATION TESTS, STUDYING FILM THICKNESS AND UNIFORMITY)

A/JONES, J. R.; B/GARDOS, M. N. B/(HUGHES AIRCRAFT CO., CULVER
CITY, CALIF.)

IN INTERNATIONAL CONFERENCE ON SOLID LUBRICATION, 1ST, DENVER,
COLO., AUGUST 24-27, 1971, PROCEEDINGS. (A72-18583 C6-15) PARK RIDGE,
ILL., AMERICAN SOCIETY OF LUBRICATION ENGINEERS, 1971, P. 185-197.
RESEARCH SPONSORED BY THE HUGHES AIRCRAFT CO.

/*FILM THICKNESS/*LUBRICANT
TESTS/*POLYIMIDES/*POLYTETRAFLUOROETHYLENE/*SELF LUBRICATING
MATERIALS/*SPACECRAFT LUBRICATION/ BALL BEARINGS/ COMPOSITE MATERIALS/
CONFERENCES/ MOLYBDENUM DISULFIDES/ PHOTOMICROGRAPHY/ SLIDING FRICTION/
THICK FILMS

71A21160 ISSUE 8 PAGE 1298 CATEGORY 15 ASLE PREPRINT 70LC-17
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FRICTION AND WEAR CHARACTERISTICS OF LUBRICATIVE COMPOSITES IN AIR
AND IN VACUUM

(FRICTION-WEAR CHARACTERISTICS OF SELF LUBRICATING COMPOSITES UNDER
SLIDING CONDITIONS IN AIR AND VACUUM)

A/GARDOS, M. N.; B/JONES, J. R. (AA/HUGHES AIRCRAFT CO., CULVER
CITY, CALIF./.) MEMBERS, \$0.75, NONMEMBERS, \$1.50.

CHICAGO, ILLINOIS, AMERICAN SOCIETY OF LUBRICATION ENGINEERS,
AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND AMERICAN SOCIETY OF
LUBRICATION ENGINEERS, LUBRICATION CONFERENCE, CINCINNATI, OHIO, OCT.
13-15, 1970.

/*COMPOSITE MATERIALS/*LUBRICANT TESTS/*SELF LUBRICATION/*SLIDING
FRICTION/*VACUUM EFFECTS/ AIR/ CONFERENCES/ SPACECRAFT LUBRICATION/
WEAR

71A21155 ISSUE 8 PAGE 1322 CATEGORY 18 ASLE PREPRINT 70LC-5
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STRUCTURAL MODIFICATIONS OF FLUORO-ALKYL S-TRIAZINES AND THEIR
LUBRICANT PROPERTIES

(FLUORO-ALKYL S-TRIAZINES AS HIGH TEMPERATURE LUBRICANTS AND ENERGY
TRANSFER FLUIDS FOR AEROSPACE SYSTEMS)

A/SNYDER, C. E., JR. (AA/USAF, MATERIALS LAB., WRIGHT- PATTERSON
AFB, OHIO/.) MEMBERS, \$0.75, NONMEMBERS, \$1.50.

CHICAGO, ILLINOIS, AMERICAN SOCIETY OF LUBRICATION ENGINEERS,
AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND AMERICAN SOCIETY OF
LUBRICATION ENGINEERS, LUBRICATION CONFERENCE, CINCINNATI, OHIO, OCT.
13-15, 1970.

/*AZINES/*ENERGY TRANSFER/*FLUORHYDROCARBONS/*HIGH TEMPERATURE
LUBRICANTS/*SPACECRAFT LUBRICATION/*WORKING FLUIDS/ AEROSPACE SYSTEMS/
CHEMICAL PROPERTIES/ CONFERENCES/ PHYSICAL PROPERTIES/ THERMAL
STABILITY

71A16174# ISSUE 5 PAGE 757 CATEGORY 15 69/00/00 4 PAGES
TRANSLATION. UNCLASSIFIED DOCUMENT

PERFORMANCE OF SOLID LUBRICANT COATINGS
(ATMOSPHERIC AND HIGH VACUUM PERFORMANCE TESTS OF SOLID LUBRICANT
COATINGS BASED ON MOLYBDENUM DISULFIDE)

A/KURILEV, G. V.

RUSSIAN ENGINEERING JOURNAL, VOL. 49, NO. 12, P. 28-31. /VESTNIK
MASHINOSTROENIYA, NO. 12, 1969./

/*INORGANIC COATINGS/*LUBRICANT TESTS/*MOLYBDENUM
DISULFIDES/*PERFORMANCE TESTS/*SOLID LUBRICANTS/ HIGH VACUUM/ HUMIDITY/
VACUUM EFFECTS

70A45862 ISSUE 24 PAGE 4295 CATEGORY 3 SAE PAPER 700781
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DESIGN CRITERIA FOR SPACECRAFT FLUID MECHANICAL SYSTEMS
(SPACECRAFT FLUID MECHANICAL SYSTEM DESIGN, CONSIDERING
ENVIRONMENTS, MATERIALS, LUBRICATION, ZERO GRAVITY AND INFLIGHT
MAINTENANCE)

A/HUDSON, J. T. (AA/MARTIN MARIETTA CORP., ADVANCED CONCEPTS
DEPT., BALTIMORE, MD./.) MEMBERS, \$1.00, NONMEMBERS, \$1.50.

NEW YORK, SOCIETY OF AUTOMOTIVE ENGINEERS, INC., SOCIETY OF
AUTOMOTIVE ENGINEERS, NATIONAL AERONAUTIC AND SPACE ENGINEERING AND
MFG. MEETING, LOS ANGELES, CALIF., OCT. 5-9, 1970.

/*FLUIDICS/*MECHANICAL DEVICES/*SPACECRAFT DESIGN/*SYSTEMS
ENGINEERING/ CONFERENCES/ SPACECRAFT CONSTRUCTION MATERIALS/ SPACECRAFT
ENVIRONMENTS/ SPACECRAFT LUBRICATION/ WEIGHTLESSNESS

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AEROSPACE LUBRICATION FOR ADVANCED VEHICLES
(LUBRICATION TECHNOLOGY FOR SPACE SHUTTLES, DISCUSSING VEHICLES AND
AIRBREATHING AND ROCKET ENGINES)

A/JOHNSON, R. L.; B/MANGANIELLO, E. J. (AB/NASA, LEWIS RESEARCH
CENTER, CLEVELAND, OHIO/.)

GERMAN SOCIETY OF TRIBOLOGY, ANNUAL MEETING, ESSEN, WEST
GERMANY, SEP. 22, 23, 1970, PAPER.

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BREATHING ENGINES/ CONFERENCES/ ROCKET ENGINES

70A35247 ISSUE 17 PAGE 3060 CATEGORY 11 69/00/00 10 PAGES
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THE PUMPING SYSTEM OF THE ULTRA-HIGH VACUUM SPACE ENVIRONMENTAL
CHAMBER

(CRYOPUMPING SYSTEMS OF ULTRAHIGH VACUUM SPACE ENVIRONMENTAL
CHAMBERS)

A/ASAI, J.; B/NAKAGAWA, H.; C/SASAKI, T.; D/TSUNODA, R.
(AC/MINISTRY OF POSTS AND TELECOMMUNICATIONS, RADIC RESEARCH
LABS., TOKYO, JAPAN/, AD/JAPAN OXYGEN CO., LTD., KAWASAKI, JAPAN/.)

TOKYO, AGNE PUBLISHING, INC., IN- INTERNATIONAL SYMPOSIUM ON
SPACE TECHNOLOGY AND SCIENCE, 8TH, TOKYO, JAPAN, AUG. 25-30, 1969,
PROCEEDINGS. P. 481-490. /A7C- 35201 17-30/

/*CRYOPUMPING/*SPACE ENVIRONMENT SIMULATION/*ULTRAHIGH
VACUUM/*VACUUM CHAMBERS/ CONFERENCES/ ION PUMPS/ LUBRICATION/ VACUUM
PUMPS

70A34166# ISSUE 16 PAGE 2923 CATEGORY 15 7C/00/00 8 PAGES
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EVALUATION OF DRY LUBRICANTS AND BEARINGS FOR SPACECRAFT
APPLICATIONS

(INSTRUMENT SIZE BALL BEARINGS LUBRICATED WITH BONDED DRY OR
TRANSFER FILMS IN SIMULATED INTERPLANETARY SPACECRAFT TESTS)

A/KIRKPATRICK, D. L.; B/YOUNG, W. C. (AB/GENERAL ELECTRIC CO.,
VALLEY FORGE SPACE TECHNOLOGY CENTER, KING OF PRUSSIA, PA./.)

SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
B - BEARINGS AND SUSPENSIONS. EDITED BY G. G. HERZL. /A7C-34154 16-15/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 107-114.

/*BALL BEARINGS/*INTERPLANETARY SPACECRAFT/*SOLID LUBRICANTS/*SPACE
ENVIRONMENT SIMULATION/*SPACECRAFT LUBRICATION/ BONDING/ PERFORMANCE
TESTS/ SERVICE LIFE/ VACUUM EFFECTS

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CONTROLLED-LEAKAGE SEALING OF BEARINGS FOR FLUID LUBRICATION IN A
SPACE VACUUM ENVIRONMENT

(CONTROLLED LEAKAGE SEALING OF HYDRODYNAMIC BEARING LLBRICATION
SYSTEMS FOR SPACE VEHICLES IN SYNCHRONOUS ORBIT)

A/SILVERSHER, H. I. (AA/LOCKHEED MISSILES AND SPACE CC.,
SUNNYVALE, CALIF./.)

SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
B - BEARINGS AND SUSPENSIONS. EDITED BY G. G. HERZL. /A70-34154 16-15/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 27-33.

/*BEARINGS/*LUBRICATION SYSTEMS/*SEALING/*SPACECRAFT
LUBRICATION/*SYNCHRONOUS SATELLITES/ LEAKAGE/ LONG TERM EFFECTS/ SPACE
MISSIONS/ VACUUM EFFECTS

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LUBRICATION OF DC MOTORS, SLIP RINGS, BEARINGS, AND GEARS FOR
LONG-LIFE SPACE APPLICATIONS

(LUBRICATION SYSTEM FLIGHT PERFORMANCE AND LABCRATORY TEST DATA FOR
SPACE APPLICATIONS, CONSIDERING TORQUE MOTORS, SLIP RINGS, BEARINGS AND
GEARS)

A/MAYER, R. W.; B/PERRIN, B. J. (AA/BALL BRCS. RESEARCH CORP.,
BOULDER, COLO./.)

SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
B - BEARINGS AND SUSPENSIONS. EDITED BY G. G. HERZL. /A70-34154 16-15/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 17-26.

/*LUBRICATION SYSTEMS/*PERFORMANCE TESTS/*SPACECRAFT LUBRICATION/
BEARINGS/ GEARS/ ORGANIC COMPOUNDS/ TORQUE MOTORS/ VACUUM EFFECTS

70A34155# ISSUE 16 PAGE 2923 CATEGORY 15 70/00/00 4 PAGES
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LUBRICATION AS PART OF TOTAL DESIGN

(LUBRICATION ROLE IN AEROSPACE ENGINEERING, DISCLSSING LUBRICANT AND
COMPONENT SELECTION, ENVIRONMENTAL FACTORS, ETC)

A/CLAUSS, F. J. (AA/LOCKHEED MISSILES AND SPACE CC., SUNNYVALE,
CALIF./.)

SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
B - BEARINGS AND SUSPENSIONS. EDITED BY G. G. HERZL. /A70-34154 16-15/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 3-6.

/*AEROSPACE ENGINEERING/*LUBRICANTS/*SPACECRAFT LUBRICATION/
AEROSPACE ENVIRONMENTS/ BALL BEARINGS/ ELECTRIC MOTCRS/ GEARS/ VACUUM
EFFECTS

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UNIQUE MECHANISM FEATURES OF ATS STABILIZATION BCM PACKAGES
(ATS ATTITUDE STABILIZATION BOOM PACKAGES, DESCRIBING TORQUE
TRANSMISSION, DRUM SYNCHRONIZATION, ELECTRICAL ISCLATION, LUBRICATION,
ETC)

A/GRIMSHAW, F. R.; B/LOHNES, R. A.; C/MATTEO, D. N. (AC/GE
VALLEY FORGE SPACE TECHNOLOGY CENTER, KING OF PRUSSIA, PA./, AA/SPAR
AEROSPACE PRODUCTS, LTD., TORONTO, CANADA/.)

SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
A - GENERAL APPLICATIONS. EDITED BY G. G. HERZL. /A70-34101 16-31/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 399-407.

/*APPLICATIONS TECHNOLOGY SATELLITES/*BOOMS (EQUIPMENT)/*SATELLITE
ATTITUDE CONTROL/ DEPLOYMENT/ GEARS/ MECHANICAL DEVICES/ SPACECRAFT
LUBRICATION/ WEIGHTLESSNESS

70A34115# ISSUE 16 PAGE 2985 CATEGORY 31 70/00/00 7 PAGES
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DESPINNING THE ATS SATELLITE
(TWO STAGE YO-YO WITH NUTATION DAMPER FOR DESPINNING ATS SYNCHRONOUS
SATELLITES, DESCRIBING MECHANICAL AND THERMAL DESIGN, LUBRICATION,
ASSEMBLY, TESTING, ETC)

A/DALLAS, J. P. (AA/HUGHES AIRCRAFT CO., LOS ANGELES, CALIF./.)
SAN FRANCISCO, J. W. STACEY, INC., IN- AEROSPACE MECHANISMS. PART
A - GENERAL APPLICATIONS. EDITED BY G. G. HERZL. /A70-34101 16-31/
/AEROSPACE MECHANISMS SERIES. VOLUME 1/, P. 139-145.

/*APPLICATIONS TECHNOLOGY SATELLITES/*SPIN REDUCTION/*SYNCHRONOUS
SATELLITES/*YO-YO DEVICES/ COST REDUCTION/ MECHANICAL DEVICES/
PERFORMANCE TESTS/ SPACECRAFT LUBRICATION/ SPACECRAFT RELIABILITY/
VACUUM EFFECTS

70A33811# ISSUE 16 PAGE 2920 CATEGORY 15 69/00/00 13 PAGES
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SLIP RING ASSEMBLIES FOR SPACECRAFT DEVICES
(SLIP RING ASSEMBLIES FOR SPACECRAFT DEVICES, EVALUATING SLIDING
ELECTRICAL CONTACT INDUSTRY TECHNOLOGICAL CAPABILITIES)

A/GLOSSBRENNER, E. W. (AA/LITTON INDUSTRIES, INC., POLY-
SCIENTIFIC DIV., BLACKSBURG, VA./.)

BLACKSBURG, VA., LITTON INDUSTRIES, INC., IN- VIRGINIA
POLYTECHNIC INST. AND LITTON INDUSTRIES, SEMINAR ON SLIDING ELECTRICAL
CONTACTS IN VACUUM AND SPACE, VIRGINIA POLYTECHNIC INST., BLACKSBURG,
VA., SEP. 30- OCT. 1, 1969, PROCEEDINGS. P. 79-90. /A70-33805 16-15/

/*ELECTRIC CONTACTS/*RING STRUCTURES/*SLIDING CONTACT/*SPACECRAFT
COMPONENTS/*SPACECRAFT LUBRICATION/ CONFERENCES/ LUBRICATION SYSTEMS/
MOLYBDENUM DISULFIDES/ NIOBIUM COMPOUNDS

70A33810# ISSUE 16 PAGE 2920 CATEGORY 15 69/00/00 8 PAGES
UNCLASSIFIED DOCUMENT

PERFORMANCE OF SOLID LUBRICATED CONTACTS FOR SPACECRAFT
(SOLID LUBRICATED CONTACTS PERFORMANCE IN NIMBUS AND OAC SPACECRAFT,
DISCUSSING SLIP RING ASSEMBLY AND NOISE PROBLEMS IN AG-GRAPHITE
BRUSHES)

A/DRABEK, S.

BLACKSBURG, VA., LITTON INDUSTRIES, INC., IN- VIRGINIA
POLYTECHNIC INST. AND LITTON INDUSTRIES, SEMINAR ON SLIDING ELECTRICAL
CONTACTS IN VACUUM AND SPACE, VIRGINIA POLYTECHNIC INST., BLACKSBURG,
VA., SEP. 30- OCT. 1, 1969, PROCEEDINGS. P. 71-77. /A70-33805 16-15/
/*ELECTRIC CONTACTS/*NIMBUS SATELLITES/*OAO/*SOLID
LUBRICANTS/*SPACECRAFT LUBRICATION/ CONFERENCES/ MOLYBDENUM DISULFIDES/
PERFORMANCE/ RING STRUCTURES/ SATELLITE INSTRUMENTS/ STAR
TRACKERS

70A33809*# ISSUE 16 PAGE 2920 CATEGORY 15 69/00/00 23 PAGES
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SLIDING ELECTRICAL CONTACTS FOR UNMANNED SCIENTIFIC SATELLITES
(SLIDING ELECTRICAL CONTACTS FOR UNMANNED SCIENTIFIC SATELLITES,
DISCUSSING NIMBUS AVCS CAMERA IRIS MOTORS)

A/BABECKI, A. J. (AA/NASA, GODDARD SPACE FLIGHT CENTER, SYSTEMS
DIV., GREENBELT, MD./.)

BLACKSBURG, VA., LITTON INDUSTRIES, INC., IN- VIRGINIA
POLYTECHNIC INST. AND LITTON INDUSTRIES, SEMINAR ON SLIDING ELECTRICAL
CONTACTS IN VACUUM AND SPACE, VIRGINIA POLYTECHNIC INST., BLACKSBURG,
VA., SEP. 30- OCT. 1, 1969, PROCEEDINGS. P. 49-70. /A70-33805 16-15/
/*ELECTRIC CONTACTS/*SCIENTIFIC SATELLITES/*SLIDING
CONTACT/*SPACECRAFT CONSTRUCTION MATERIALS/ CAMERA SHUTTERS/
CONFERENCES/ NIMBUS SATELLITES/ OAO/ SATELLITE INSTRUMENTS/ SPACECRAFT
LUBRICATION/ STAR TRACKERS

70A29120# ISSUE 13 PAGE 2419 CATEGORY 15 70/05/00 6 PAGES
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LUBRICANTS FOR SPACE. II

(VACUUM EVALUATION OF LUBRICANTS AND TECHNIQUES APPLICABLE TO
MINIATURE BALL BEARINGS AND INSTRUMENT BALL BEARINGS AND INSTRUMENT
GEARING FOR SPACE SYSTEMS)

A/HARRIS, C. L. (AA/ELLIOTT BROS. /LONDON/, LTD., ELLIOTT SPACE
AND WEAPONS RESEARCH LAB., FRIMLEY, SURREY, ENGLAND/.)

MECHANICAL ENGINEERING, VOL. 92, P. 48-53. NEW YORK, RESEARCH
SUPPORTED BY THE MINISTRY OF TECHNOLOGY. /AMERICAN SOCIETY OF
LUBRICATION ENGINEERS AND AMERICAN SOCIETY OF MECHANICAL ENGINEERS,
JOINT LUBRICATION CONFERENCE, HOUSTON, TEX., OCT. 13-16, 1969./

/*BALL BEARINGS/*GEARS/*SPACECRAFT LUBRICATION/*VACUUM EFFECTS/
CONFERENCES/ LOADS (FORCES)/ SELF LUBRICATION/ TORQUE/ VIBRATION
EFFECTS

70A17780# ISSUE 6 PAGE 1141 CATEGORY 30 69/CO/00 6 PAGES IN
FRENCH IN FRENCH. UNCLASSIFIED DOCUMENT
DESCRIPTION OF SOME IMPORTANT APPLIED RESEARCH IN THE SPACE FIELD
(APPLIED SPACE RESEARCH COVERING PHOTOVOLTAIC CELLS, ELECTRIC
PROPULSION, ONBOARD DATA STORAGE, VACUUM LUBRICATION AND NASA-EUROPEAN
ORGANIZATIONS BUDGETS)

A/BOCHET, J.-C. (AA/ESRO, PARIS, FRANCE/.)

SCIENCES ET INDUSTRIES SPATIALES, VOL. 5, NO. 7-8, P. 19-23, 51.

/*BUDGETING/*DATA STORAGE/*ELECTRIC PROPULSION/*PHOTOVOLTAIC
CELLS/*SPACECRAFT LUBRICATION/ AEROSPACE INDUSTRY/ ENERGY CONVERSION/
EUROPEAN SPACE PROGRAMS/ NASA PROGRAMS/ ONBOARD EQUIPMENT/ TECHNOLOGY
UTILIZATION

70A17340 ISSUE 6 PAGE 1075 CATEGORY 15 69/CC/00 10 PAGES
UNCLASSIFIED DOCUMENT

TESTING SOLID LUBRICANTS

(SOLID LUBRICANTS FRICTION AND WEAR BENCH TESTING UNDER SIMULATED
SPACE ENVIRONMENT)

A/AZZAM, H. T. (AA/DOW CORNING CORP., TRUMBULL, CONN./.)

NORTH HOLLYWOOD, CALIF., WESTERN PERIODICALS CO., CONVENTION
SPONSORED BY THE INST. OF ELECTRICAL AND ELECTRONICS ENGINEERS AND THE
WESTERN ELECTRONIC MANUFACTURERS ASSN. /IECP SYMPOSIUM RECORD. VOLUME
10/, DATE- 1969. IN- WESTERN ELECTRONIC SHOW AND CONVENTION,
INTERNATIONAL ELECTRONIC CIRCUIT PACKAGING SYMPOSIUM, SAN FRANCISCO,
CALIF., AUG. 20, 21, 1969, PROCEEDINGS. PART 7. P. 3/2 1-3/2 10.
/A70-17334 06-09/

/*SOLID LUBRICANTS/*SPACE ENVIRONMENT SIMULATION/*SPACECRAFT
LUBRICATION/*WEAR TESTS/ CONFERENCES/ DEEP SPACE/ SERVICE LIFE

70A11693# ISSUE 2 PAGE 360 CATEGORY 30 69/CC/00 288 PAGES IN
RUSSIAN IN RUSSIAN. UNCLASSIFIED DOCUMENT

PHYSICOTECHNOLOGICAL BASIS OF SPACE RESEARCH

(SOVIET BOOK ON PHYSICOTECHNOLOGICAL BASIS OF SPACE RESEARCH
COVERING NEAR EARTH AND INTERPLANETARY ENVIRONMENTAL FACTORS AND
EFFECTS ON SPACECRAFT DESIGNS AND MATERIALS)

A/KROSHKIN, M. G.

MOSCOW, IZDATEL'STVO MASHINOSTROENIE,

/*AEROSPACE ENVIRONMENTS/*AEROSPACE SCIENCES/*INTERPLANETARY
SPACE/*SPACECRAFT DESIGN/ AEROSPACE INDUSTRY/ EARTH ATMOSPHERE/
RADIATION DAMAGE/ SPACECRAFT COMPONENTS/ SPACECRAFT LUBRICATION/ VACUUM
EFFECTS

70A10379# ISSUE 1 PAGE 100 CATEGORY 15 ASME PAPER 69-LUB-30
69/10/00 13 PAGES UNCLASSIFIED DOCUMENT

VACUUM EVALUATION OF LUBRICANTS AND TECHNIQUES FOR SPACE-EXPOSED COMPONENTS

(VACUUM EVALUATION OF LUBRICANTS AND TECHNIQUES APPLICABLE TO MINIATURE BALL BEARINGS AND INSTRUMENT GEARING FOR SPACE SYSTEMS)

A/HARRIS, C. L. (AA/ELLIOTT BROTHERS /LONDON/, LTD., ELLIOTT SPACE AND WEAPONS RESEARCH LAB., FRIMLEY, SURREY, ENGLAND/.) MEMBERS, \$0.75, NONMEMBERS, \$1.50.

NEW YORK, AMERICAN SOCIETY OF MECHANICAL ENGINEERS, RESEARCH SUPPORTED BY THE MINISTRY OF TECHNOLOGY. AMERICAN SOCIETY OF LUBRICATION ENGINEERS AND AMERICAN SOCIETY OF MECHANICAL ENGINEERS, JOINT LUBRICATION CONFERENCE, HOUSTON, TEX., OCT. 13-16, 1969.

/*BALL BEARINGS/*GEARS/*SPACECRAFT LUBRICATION/*VACUUM EFFECTS/ CONFERENCES/ LOADS (FORCES)/ SELF LUBRICATION/ TORQUE/ VIBRATION EFFECTS

69A26893# ISSUE 12 PAGE 2175 CATEGORY 31 69/00/00 5 PAGES UNCLASSIFIED DOCUMENT

A FLYWHEEL FOR STABILIZING SPACE VEHICLES.

(SPACE VEHICLE ATTITUDE STABILIZATION BASED ON ROTATING FLYWHEEL RESISTANCE TO CHANGES IN ROTATION AXIS ATTITUDE, USING GREASE LUBRICATED HYDRODYNAMIC BEARING SYSTEM)

A/REINHOUDT, J. P. (AA/PHILIPS' GLOEILAMPENFABRIKEN, PHILIPS RESEARCH LABS., EINDHOVEN, NETHERLANDS/.)

PHILIPS TECHNICAL REVIEW, VOL. 30, NO. 1, P. 2-6.

/*ATTITUDE STABILITY/*FLYWHEELS/*SATELLITE ORIENTATION/*SPACECRAFT STABILITY/ ANTIFRICTION BEARINGS/ GREASES/ JOURNAL BEARINGS/ SPACECRAFT LUBRICATION/ STABILIZATION

69A18564 ISSUE 7 PAGE 1138 CATEGORY 15 IME PAPER 10 69/01/00 8 PAGES UNCLASSIFIED DOCUMENT

A SHORT SURVEY OF EUROPEAN WORK ON LUBRICATION IN VACUUM.

(EUROPEAN RESEARCH ON ULTRAHIGH VACUUM LUBRICATION IN SPACE ENVIRONMENTS, EMPHASIZING FRICTION OF MATERIALS UNDER VARIOUS LOADINGS AND TEMPERATURES)

A/BRISCOE, H. M.; B/DAUPHIN, J. (AB/EUROPEAN SPACE RESEARCH ORGANIZATION, EUROPEAN SPACE RESEARCH AND TECHNOLOGY CENTRE, NOORDWIJK, NETHERLANDS/.) \$7.20 PER SET OF 12.

LONDON, ENGLAND. INSTITUTION OF MECHANICAL ENGINEERS, INSTITUTION OF MECHANICAL ENGINEERS, SYMPOSIUM ON LUBRICATION IN HOSTILE ENVIRONMENTS, LONDON, ENGLAND, JAN. 15, 16, 1969.

/*AEROSPACE ENVIRONMENTS/*EUROPEAN SPACE PROGRAMS/*FRICTION/*SPACECRAFT LUBRICATION/*ULTRAHIGH VACUUM/ BEARINGS/ CONFERENCES/ LOAD TESTS/ TEMPERATURE EFFECTS

69A18562 ISSUE 7 PAGE 1138 CATEGORY 15 IME PAPER 6 69/01/00
13 PAGES UNCLASSIFIED DOCUMENT

LUBRICATION OF BEARINGS AND GEARS FOR OPERATION IN A SPACE ENVIRONMENT.

(MINIATURE BALL AND JEWEL BEARINGS AND GEAR LUBRICATION IN ULTRAHIGH VACUUM TESTS FOR SPACE ENVIRONMENT OPERATION)

A/HARRIS, C. L.; B/WARWICK, M. G. (AB/ELLIOTT-AUTOMATIC, LTD., SPACE AND WEAPONS RESEARCH LAB., CAMBERLEY, SURREY, ENGLAND/.) \$7.20 PER SET OF 12.

LONDON, ENGLAND, INSTITUTION OF MECHANICAL ENGINEERS, RESEARCH SUPPORTED BY THE MINISTRY OF TECHNOLOGY. INSTITUTION OF MECHANICAL ENGINEERS, SYMPOSIUM ON LUBRICATION IN HOSTILE ENVIRONMENTS, LONDON, ENGLAND, JAN. 15, 16, 1969.

/*ANTIFRICTION BEARINGS/*GEARS/*LUBRICATION/*SPACE ENVIRONMENT SIMULATION/*WEAR TESTS/ BALL BEARINGS/ BONDING/ COMPOSITE MATERIALS/ CONFERENCES/ FRICTION REDUCTION/ LIFE (DURABILITY)/ POROUS MATERIALS/ SOLID LUBRICANTS/ ULTRAHIGH VACUUM/ VAPOR PRESSURE

69A18561 ISSUE 7 PAGE 1138 CATEGORY 15 IME PAPER 7 69/01/00
13 PAGES UNCLASSIFIED DOCUMENT

FRICTION AND WEAR STUDIES IN ULTRA-HIGH VACUUM AND THE EVALUATION OF ELECTRICAL SLIP-RINGS.

(DRY SLIDING FRICTION AND WEAR IN ULTRAHIGH VACUUM, EMPHASIZING SLIP RINGS AND BRUSHES FOR SPACE APPLICATIONS)

A/HARRIS, C. L.; B/WYN-ROBERTS, D. (AB/ELLIOTT-AUTOMATIC, LTD., SPACE AND WEAPONS RESEARCH LAB., CAMBERLEY, SURREY, ENGLAND/.) \$7.20 PER SET OF 12.

LONDON, ENGLAND. INSTITUTION OF MECHANICAL ENGINEERS, INSTITUTION OF MECHANICAL ENGINEERS, SYMPOSIUM ON LUBRICATION IN HOSTILE ENVIRONMENTS, LONDON, ENGLAND, JAN. 15, 16, 1969.

/*DRY FRICTION/*ELECTRIC CONTACTS/*SLIDING FRICTION/*ULTRAHIGH VACUUM/*WEAR TESTS/ BRUSHES/ CONFERENCES/ FRICTION REDUCTION/ GEARS/ LUBRICATION/ SPACE ENVIRONMENT SIMULATION/ STATIC FRICTION/ VAPOR PRESSURE

69A17046# ISSUE 6 PAGE 945 CATEGORY 18 68/08/00 8 PAGES
UNCLASSIFIED DOCUMENT

ACHIEVEMENTS IN STUDIES OF PROPERTIES OF MATERIALS USED FOR SPACE OBJECTS AND THEIR USE IN INDUSTRY.

(SATELLITE, SPACECRAFT AND ROCKET COMPONENTS COATINGS AND LUBRICANTS, NOTING HIGH RADIATION RESISTANCE OF POLYIMIDES)

A/KONRADI, G. G.; B/KOZELKIN, V. V.; C/NOVITSKII, L. A.; D/SHVAREV, V. V.

UNITED NATIONS, CONFERENCE ON THE EXPLORATION AND PEACEFUL USES OF OUTER SPACE, VIENNA, AUSTRIA, AUG. 14-27, 1968, PAPER.

/*POLYIMIDES/*PROTECTIVE COATINGS/*RADIATION TOLERANCE/*SPACECRAFT LUBRICATION/ CONFERENCES/ POLYTETRAFLUOROETHYLENE/ POLYVINYL CHLORIDE/ TECHNOLOGY UTILIZATION/ U.S.S.R. SPACE PROGRAM

69A11766 ISSUE 2 PAGE 252 CATEGORY 15 68/00/00 18 PAGES
UNCLASSIFIED DOCUMENT

SPACE MECHANISM LUBRICATION.

(SPACE LUBRICATION SYSTEM FOR ORBITING SOLAR OBSERVATORY PROGRAM,
DISCUSSING THEORETICAL HIGH VACUUM PRINCIPLES AND FLIGHT PERFORMANCE
AND ENVIRONMENTAL TEST DATA)

A/MAYER, R. W.; B/PERRIN, B. J. (AA/BALL BROTHERS RESEARCH CORP.,
BOULDER, COLO./.)

TARZANA, CALIF., AMERICAN ASTRONAUTICAL SOCIETY, SYMPOSIUM
SPONSORED BY THE AMERICAN ASTRONAUTICAL SOCIETY AND THE AMERICAN INST.
OF AERONAUTICS AND ASTRONAUTICS. IN- SPACE PROJECTIONS FROM THE ROCKY
MOUNTAIN REGION, PROCEEDINGS OF THE SYMPOSIUM, DENVER, COLO., JUL. 15,
16, 1968. VOLUME 3. <A69-11755 02-11<

/*ENVIRONMENTAL TESTS/*LUBRICATION SYSTEMS/*OSC/*SPACECRAFT
LUBRICATION/*VACUUM EFFECTS/ CONFERENCES/ GROUND TESTS/ HIGH VACUUM/
ORGANIC LIQUIDS/ PERFORMANCE TESTS/ SPACE PROGRAMS

68A43941 ISSUE 23 PAGE 4387 CATEGORY 11 68/00/00 7 PAGES
UNCLASSIFIED DOCUMENT

A MULTIPLE ENVIRONMENT TEST SYSTEM FOR COMBINED ENVIRONMENT SPACE
SIMULATION.

(FLEXIBLE SYSTEM FOR ENVIRONMENT TESTING OF INSTRUMENT BEARING
LUBRICANTS)

A/TIERNEY, W. L. (AA/LOCKHEED AIRCRAFT CORP., LOCKHEED MISSILES
AND SPACE CO., SUNNYVALE, CALIF./.)

NORTH HOLLYWOOD, CALIF., WESTERN PERIODICALS CO. /SYMPOSIUM
RECORD. VOLUME 9/, CONVENTION SPONSORED BY THE INST. OF ELECTRICAL AND
ELECTRONICS ENGINEERS AND THE WESTERN ELECTRONIC MANUFACTURERS ASSN.
IN- WESTERN ELECTRONIC SHOW AND CONVENTION, INTERNATIONAL ELECTRONIC
CIRCUIT PACKAGING SYMPOSIUM, LOS ANGELES, CALIF., AUG. 19, 20, 1968,
PROCEEDINGS. <A68-43935 23-09< P. 4/3-1 TO 4/3-7.

/*ELECTRONIC EQUIPMENT TESTS/*ENVIRONMENTAL TESTS/*SPACE
ENVIRONMENT SIMULATION/*TEST FACILITIES/ BEARINGS/ CONFERENCES/
HARDWARE/ LUBRICANTS/ SYSTEMS ENGINEERING/ TABLES (DATA)

68A38760* ISSUE 20 PAGE 3779 CATEGORY 15 68/00/00 14 PAGES
IN FRENCH IN FRENCH. UNCLASSIFIED DOCUMENT

PROBLEMS OF LUBRICATION AND BEARINGS IN SPACE VACUUM
(LUBRICATION AND BEARINGS IN SPACE VACUUM ENVIRONMENT, EXAMINING
GREASE, LIQUID AND SEMISOLIDS EVAPORATION RATE BY LANGMUIR EQUATION)
A/BISSON, E. E. (AA/NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO/
COURTEL, R. /CENTRE NATIONALE DE LA RECHERCHE SCIENTIFIQUE, BELLEVUE,
HAUTS-DE-SEINE, FRANCE/.)

PARIS, INSTITUT FRANCAIS DES COMBUSTIBLES ET DE L'ENERGIE, IN-
STATES OF MATTER UNDER THE EXTREME EFFECTS OF VERY HIGH AND VERY LOW
TEMPERATURES, AND VERY HIGH AND VERY LOW PRESSURES, INSTITUT FRANCAIS
DES COMBUSTIBLES ET DE L'ENERGIE, INTERNATIONAL CONFERENCE, 7TH, PARIS,
FRANCE, APR. 4-8, 1967, PROCEEDINGS (ETATS DE LA MATIERE SOUS LES EFFETS
EXTREMES DES TRES HAUTES ET TRES BASSES TEMPERATURES, TRES HAUTES ET
TRES BASSES PRESSIONS, INSTITUT FRANCAIS DES COMBUSTIBLES ET DE
L'ENERGIE, JOURNEES INTERNATIONALES, 7TH, PARIS, FRANCE, APR. 4-8,
1967, PROCEEDINGS). (A68-38739 20-23) P. 529-542.

/*AEROSPACE ENVIRONMENTS/*BALL BEARINGS/*EVAPORATION
RATE/*SPACECRAFT LUBRICATION/*VACUUM EFFECTS/ CONFERENCES/ LUBRICATING
OILS/ SOLID LUBRICANTS

68A34571 ISSUE 17 PAGE 3176 CATEGORY 15 66/11/00 8 PAGES IN
FRENCH IN FRENCH. UNCLASSIFIED DOCUMENT

FRICITION IN VACUUM

(FRICTION CHARACTERISTICS IN VACUUM AND ULTRAHIGH VACUUM, DISCUSSING
SOLID AND SELF LUBRICATING MATERIALS FOR SPACE APPLICATIONS)

A/COURTEL, R. (AA/CENTRE NATIONAL DE LA RECHERCHE SCIENTIFIQUE,
CHATILLON-SOUS-BAGNEUX, HAUTS-DE-SEINE, FRANCE/.)

ANNALES DES MINES, P. 1084-1091. 21 /CENTRE NATIONAL D'ETUDES
SPATIALES, COLLOQUE INTERNATIONAL SUR L'ELABORATION ET L'EMPLOI DES
MATERIAUX RESISTANTS POUR LES APPLICATIONS SPATIALES, PARIS, FRANCE,
FEB. 7- 11, 1966./

/*SELF LUBRICATING MATERIALS/*SLIDING FRICTION/*SOLID
LUBRICANTS/*SPACECRAFT LUBRICATION/*ULTRAHIGH VACUUM/*VACUUM EFFECTS/
CONFERENCES/ METALLURGY/ SPACECRAFT DESIGN

68A34169 ISSUE 17 PAGE 3174 CATEGORY 15 67/00/00 14 PAGES
UNCLASSIFIED DOCUMENT

SOME ASPECTS OF THE SURFACE CHEMISTRY OF ADHESION AND OF FRICTION.
(SURFACE CHEMISTRY ADHESION AND RELATION TO FRICTION NOTING METHODS
FOR REDUCTION IN HIGH VACUUM ENVIRONMENTS)

A/ADAMSON, A. W. (AA/SOUTHERN CALIFORNIA, U., DEPT. OF CHEMISTRY,
LOS ANGELES, CALIF./.)

PHILADELPHIA, AMERICAN SOCIETY FOR TESTING AND MATERIALS /ASTM
SPECIAL TECHNICAL PUBLICATION NO. 431/, IN- ADHESION OR COLD WELDING
OF MATERIALS IN SPACE ENVIRONMENTS, AMERICAN SOCIETY FOR TESTING AND
MATERIALS AND AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL
MEETING, TORONTO, CANADA, MAY 1, 2, 1967, PROCEEDINGS. <A68-34168
17-15< P. 6-19.

/*ADHESION/*AEROSPACE ENVIRONMENTS/*FRICTION REDUCTION/*HIGH
VACUUM/*SURFACE REACTIONS/ CONFERENCES/ LIQUIDS/ LUBRICANTS/ SOLIDS/
SURFACE PROPERTIES

68A30318# ISSUE 14 PAGE 2608 CATEGORY 15 AICHE PAPER 180
68/03/00 26 PAGES UNCLASSIFIED DOCUMENT

LUBRICANTS FOR THE SPACE ENVIRONMENT.

(SPACE ENVIRONMENT LUBRICANT SELECTION FOR SPECIFIC SPACE SYSTEMS
AND APPLICATIONS, GIVING TABLES OF LUBRICANTS FOR VARIOUS SATELLITES)

A/FLOM, D. G.; B/HALTNER, A. J. (AB/GENERAL ELECTRIC CO.,
AEROSPACE GROUP, MISSILE AND SPACE DIV., SPACE SCIENCES LAB., VALLEY
FORGE, PA./.) \$0.50.

NEW YORK, AMERICAN INST. OF CHEMICAL ENGINEERS, AMERICAN INST.
OF CHEMICAL ENGINEERS, MATERIALS CONFERENCE, SYMPOSIUM ON MATERIALS FOR
RE-ENTRY AND SPACECRAFT SYSTEMS - SPACECRAFT MATERIALS, PART 2,
PHILADELPHIA, PA., MAR. 31-APR. 4, 1968.

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ENVIRONMENTS/ AEROSPACE SYSTEMS/ CHEMICAL ENGINEERING/ CONFERENCES/
GRAPHITE/ SYSTEMS ENGINEERING/ TABLES (DATA)/ VACUUM SYSTEMS

68A29990* ISSUE 14 PAGE 2608 CATEGORY 15 JPL-950056 NAS7-110
68/05/00 13 PAGES UNCLASSIFIED DOCUMENT

FRICTION AND WEAR OF POTENTIOMETER CONTACTS IN VACUUM.

(FRICTION AND WEAR OF WIRE WOUND POTENTIOMETERS SLIDING ELECTRIC
CONTACTS IN VACUUM, EVALUATING LUBRICATIVE POWDERS AND CONDUCTIVE
COMPOSITES)

A/JONES, J. R. (AA/HUGHES AIRCRAFT CO., CULVER CITY, CALIF./.)
WEAR, VOL. 11, P. 355-367.

/*ELECTRIC CONTACTS/*POTENTIOMETERS (INSTRUMENTS)/*SLIDING
FRICTION/*SOLID LUBRICANTS/*VACUUM EFFECTS/*WEAR TESTS/ FRICTION
REDUCTION/ METAL FILMS/ METAL POWDER/ SELF LUBRICATING MATERIALS/
SLIDING CONTACT/ SPACECRAFT LUBRICATION

68A27386# ISSUE 12 PAGE 2228 CATEGORY 15 ASME PAPER 68-DE-39
68/04/00 9 PAGES UNCLASSIFIED DOCUMENT

LUBRICATION TECHNIQUES FOR USE IN VACUUM.

(LUBRICATION TECHNIQUES FOR VACUUM ENVIRONMENTS OF SPACE VEHICLES
INCLUDING OILS, GREASES AND SOLID LUBRICANTS)

A/BENZING, R. J.; B/MCCONNELL, B. D. (AA/USAF, SYSTEMS COMMAND,
RESEARCH AND TECHNOLOGY DIV., MATERIALS LAB., WRIGHT-PATTERSON AFB,
OHIO/ AB/USAF, SYSTEMS COMMAND, RESEARCH AND TECHNOLOGY DIV.,
WRIGHT-PATTERSON AFB, OHIO/.)

NEW YORK, AMERICAN SOCIETY OF MECHANICAL ENGINEERS, AMERICAN
SOCIETY OF MECHANICAL ENGINEERS, DESIGN ENGINEERING CONFERENCE AND
SHOW, CHICAGO, ILL., APR. 22-25, 1968.

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ENVIRONMENTS/*SPACECRAFT LUBRICATION/ CONFERENCES/ GREASES/ LUBRICATION
SYSTEMS/ VACUUM EFFECTS

68A26726 ISSUE 12 PAGE 2226 CATEGORY 15 68/04/00 7 PAGES
UNCLASSIFIED DOCUMENT

LUBRICATION IN SPACE VACUUM. III - LIFE TEST EVALUATION OF BALL
BEARINGS LUBRICATED WITH OILS AND GREASES.

(LIFE TESTS OF BALL BEARINGS LUBRICATED WITH OILS AND GREASES DURING
OPERATION IN INDUCTION MOTORS UNDER SIMULATED SPACE VACUUM CONDITIONS)

A/HARRIS, C. L.; B/READ, J. E.; C/THOMPSON, J. B. (AC/ELLIOTT
BROTHERS /LONDON/, LTD., SPACE AND WEAPONS RESEARCH LAB., FRIMLEY,
SURREY, ENGLAND/.)

LUBRICATION ENGINEERING, VOL. 24, P. 182-188. RESEARCH SUPPORTED
BY THE MINISTRY OF AVIATION.

/*BALL BEARINGS/*LUBRICATING OILS/*SERVICE LIFE/*SPACE ENVIRONMENT
SIMULATION/*VACUUM EFFECTS/ DEGASSING/ ELECTRIC MOTORS/ ENVIRONMENTAL
ENGINEERING/ ENVIRONMENTAL TESTS/ TORQUE

68A26725* ISSUE 12 PAGE 2226 CATEGORY 15 68/04/00 10 PAGES
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EVALUATION OF SPACE LUBRICANTS UNDER OSCILLATORY AND SLOW SPEED
ROTARY MOTION.

(SOLID FILM SPACE LUBRICANT METHODS EVALUATED FOR FRICTION SURFACES
UNDER OSCILLATORY AND SLOW SPEED ROTARY MOTION)

A/VEST, C. E.; B/WARD, B. W., JR. (AB/NASA, GODDARD SPACE FLIGHT
CENTER, GREENBELT, MD./.)

LUBRICATION ENGINEERING, VOL. 24, P. 163-172. /AMERICAN SOCIETY
OF LUBRICATION ENGINEERS, ANNUAL MEETING, 22ND, TORONTO, CANADA, MAY
1-4, 1967./

/*FRICTION MEASUREMENT/*ROTATING BODIES/*SOLID
LUBRICANTS/*SPACECRAFT LUBRICATION/*STRUCTURAL VIBRATION/ BEARINGS/
CONFERENCES/ GOLD COATINGS/ MOLYBDENUM DISULFIDES/
POLYTETRAFLUOROETHYLENE

68A24572 ISSUE 10 PAGE 1784 CATEGORY 11 68/02/00 2 PAGES IN
GERMAN IN GERMAN. UNCLASSIFIED DOCUMENT
TEST STAND FOR THE MECHANICAL-DYNAMICAL TESTING OF LUBRICANTS UNDER
HIGH- AND ULTRAHIGH-VACUUM CONDITIONS
(TEST STAND FOR CHECKING LUBRICATION OF AIRCRAFT AND SPACECRAFT
PARTS UNDER FLIGHT STRESS, WEIGHTLESSNESS AND ENVIRONMENTAL CONDITIONS)
A/SPENGLER, G. (AA/DEUTSCHE VERSUCHSANSTALT FUER LUFT- UND
RAUMFAHRT, INSTITUT FUER FLUGTREIB- UND SCHMIERSTOFFE, MÜNICH, WEST
GERMANY/.)

DVL-NACHRICHTEN, P. 324, 325.
/*AIRCRAFT PARTS/*ENVIRONMENT SIMULATION/*LUBRICATION
SYSTEMS/*SPACE FLIGHT STRESS/*SPACECRAFT COMPONENTS/*TEST STANDS/
TEMPERATURE CONTROL/ ULTRAHIGH VACUUM/ WEIGHTLESSNESS

68A21757 ISSUE 9 PAGE 1596 CATEGORY 15 68/02/00 7 PAGES
UNCLASSIFIED DOCUMENT

LUBRICATION IN SPACE VACUUM /10 SUPER -10 TORR/. 1 - TESTING
TECHNIQUES FOR BALL BEARINGS.

(MODULAR TEST CAPSULE TECHNIQUE FOR VACUUM TESTING OF LIFETIMES AND
BEARING TORQUE PERFORMANCE OF DRY FILM AND OIL LUBRICATED BALL
BEARINGS)

A/HARRIS, C. L.; B/READ, J. E.; C/THOMPSON, J. B.; D/WILSON, C.
I. (AD/ELLIOTT BROTHERS /LONDON/, LTD., SPACE AND WEAPONS RESEARCH
LAB., FRIMLEY, SURREY, ENGLAND/.)

LUBRICATION ENGINEERING, VOL. 24, P. 57-63. RESEARCH SUPPORTED
BY THE MINISTRY OF AVIATION.

/*BALL BEARINGS/*LUBRICATING OILS/*MATERIALS TESTS/*TEST
EQUIPMENT/*VACUUM CHAMBERS/ BEARING ALLOYS/ ENVIRONMENTAL TESTS/ LIFE
(DURABILITY)/ LUBRICATION/ MOLECULAR FLOW/ POLYBENZYL DISULFIDES/ SPACE
ENVIRONMENT SIMULATION/ VACUUM APPARATUS

67A38142* ISSUE 21 PAGE 3633 CATEGORY 15 67/00/00 12 PAGES
UNCLASSIFIED DOCUMENT

LUBRICATION AND WEAR FUNDAMENTALS FOR HIGH- VACUUM APPLICATIONS.
(LUBRICATION AND WEAR IN HIGH VACUUM, CONSIDERING INABILITY TO
MAINTAIN OXIDE FILMS, EVAPORATION OF LUBRICANTS, HEAT TRANSFER AND
SLIDING FRICTION)

A/BUCKLEY, D. H.; B/JOHNSON, R. L. (AB/NASA, LEWIS RESEARCH
CENTER, LUBRICATION BRANCH, CLEVELAND, OHIO/ AA/NASA, LEWIS RESEARCH
CENTER, LUBRICATION BRANCH, SPACE ENVIRONMENT LUBRICATION SECTION,
CLEVELAND, OHIO/.)

IN- CONFERENCE ON LUBRICATION AND WEAR- FUNDAMENTALS AND
APPLICATION TO DESIGN, LONDON, ENGLAND, SEP. 25-29, 1967, PAPERS.
SESSION 5 - SPECIFIC ENVIRONMENTS. <A67-38140 21-15> LONDON,
INSTITUTION OF MECHANICAL ENGINEERS, 1967, P. 41-52. 31 REFS.

/*HEAT TRANSFER/*HIGH VACUUM/*LUBRICATION/*SLIDING FRICTION/*WEAR/
ADHESION/ ALLOY/ BONDING/ CONFERENCE/ CRYSTAL/ DIFFUSION/ EVAPORATION/
FILM/ FRICTION/ METAL/ OXIDE/ SHEAR/ SLIDING/ STRUCTURE

67A38140 ISSUE 21 PAGE 3633 CATEGORY 15 67/CO/00 91 PAGES
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CONFERENCE ON LUBRICATION AND WEAR- FUNDAMENTALS AND APPLICATION TO
DESIGN, LONDON, ENGLAND, SEPTEMBER 25-29, 1967, PAPERS. SESSION 5 -
SPECIFIC ENVIRONMENTS.

(LUBRICATION AND WEAR - IME CONFERENCE, LONDON, SEPTEMBER 1967,
SESSION 5, SPECIFIC ENVIRONMENTS)

PRICE OF SIX VOLUMES, \$28.50.

LONDON, INSTITUTION OF MECHANICAL ENGINEERS, 1967. 91 P.

/*CONFERENCE/*LUBRICATION/*WEAR/ BEARING/ CONTACT/ ENVIRONMENT/
HIGH VACUUM/ ROLLING/ SPACE

67A34993# ISSUE 19 PAGE 3248 CATEGORY 18 67/00/00 7 PAGES
UNCLASSIFIED DOCUMENT

LUBRICATING GREASES FOR AERONAUTICAL/AEROSPACE APPLICATIONS.

(LUBRICANT REQUIREMENTS FOR AIRCRAFT PARTS NOTING HIGH AND LOW
TEMPERATURE, RADIATION CONDITIONS, LUBRICATION CAPABILITY LIMITS, ETC)
A/NEW, L. D. (AA/NATIONAL RESEARCH COUNCIL OF CANADA, DIV. OF

MECHANICAL ENGINEERING, FUELS AND LUBRICANTS LAB., OTTAWA, CANADA/.)

CANADA, NATIONAL RESEARCH COUNCIL, DIV. OF MECHANICAL ENGINEERING
AND NATIONAL AERONAUTICAL ESTABLISHMENT, QUARTERLY BULLETIN, NO. 1,
1967, P. 73-79. 11 REFS.

/*AEROSPACE TECHNOLOGY/*LUBRICANT/*RADIATION EFFECT/*SPACE
ENVIRONMENTAL LUBRICATION/ AEROSPACE/ AIRCRAFT/ BEARING/ CONTROL/
EFFECT/ ENVIRONMENT/ EXPOSURE/ GEAR/ GREASE/ LUBRICATION/ RADIATION/
RELIABILITY/ SPACE/ SYSTEM/ TECHNOLOGY/ TEMPERATURE

67A29534 ISSUE 15 PAGE 2505 CATEGORY 18 67/CO/00 321 PAGES
UNCLASSIFIED DOCUMENT

THE EFFECTS OF THE SPACE ENVIRONMENT ON MATERIALS, SOCIETY OF
AEROSPACE MATERIAL AND PROCESS ENGINEERS, NATIONAL SYMPOSIUM AND
EXHIBIT, 11TH, ST. LOUIS, MO., APRIL 19-21, 1967, PROCEEDINGS.

(EFFECTS OF SPACE ENVIRONMENT ON MATERIALS - CONFERENCE, ST. LOUIS,
APRIL 1967) \$15.

NORTH HOLLYWOOD, CALIF., WESTERN PERIODICALS CO. /SCIENCE OF
ADVANCED MATERIALS AND PROCESS ENGINEERING SERIES. VOLUME 11/, 1967,
321 P.

/*CONFERENCE/*MATERIALS SCIENCE/*SPACE ENVIRONMENT/ DEFINITION/
ENVIRONMENT/ LUBRICATION/ MATERIAL/ SCIENCE/ SIMULATION/ SPACE/
STRUCTURAL

67A22421 ISSUE 9 PAGE 1506 CATEGORY 15 ASLE PAPER 66AM 3C2
67/02/00 5 PAGES UNCLASSIFIED DOCUMENT

ADVANCED AFROSPACE GREASES.

(GREASE LUBRICANTS FOR AEROSPACE APPLICATION, DETERMINING PHYSICAL PROPERTIES AND TESTING THEM AT 400 DEGREES F AND UNDER HIGH VACUUM)

A/BUNTING, K. R.; B/CHRISTIAN, J. B. (AB/USAF, SYSTEMS COMMAND, RESEARCH AND TECHNOLOGY DIV., MATERIALS LAB., WRIGHT-PATTERSON AFB, OHIO/ AA/AMERICAN OIL CO., WHITING, IND./.)

/AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 21ST, PITTSBURGH, PA., MAY 2-5, 1966, PAPER 66AM 3C2./ LUBRICATION ENGINEERING, VOL. 23, FEB. 1967, P. 52-56. <FOR ABSTRACT SEE ISSUE 16, PAGE 2711, ACCESSION NO. A66-30409<

/*GREASE/*LUBRICANT/*PHYSICAL PROPERTY/*SPACE ENVIRONMENTAL LUBRICATION/ AEROSPACE/ BEARING/ CONFERENCE/ ENVIRONMENT/ LOAD/ LUBRICATION/ PHYSICAL/ PRESSURE/ PROPERTY/ SPACE/ SPEED/ TECHNOLOGY/ TEMPERATURE/ TIME/ VACUUM

67A16932*# ISSUE 5 PAGE 811 CATEGORY 15 66/12/00 32 PAGES
UNCLASSIFIED DOCUMENT

LUBRICATION AND BEARING PROBLEMS IN THE VACUUM OF SPACE.

(LUBRICATION AND BEARING PROBLEMS IN SPACE ENVIRONMENT, ACTING MOLYBDENUM DISULFIDE AND SILICONES)

A/BISSON, E. E. (AA/NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO/.)

GROUPEMENT POUR L'AVANCEMENT DE LA MECANIQUE INDUSTRIELLE, JOURNEES D'ETUDE DU FROTTEMENT ET DE L'USURE, PARIS, FRANCE, DEC. 5, 6, 1966, PAPER. 32 P. 25 REFS. TRANSLATION.

/*BEARING/*MOLYBDENUM SULFIDE/*SILICONE/*SPACE ENVIRONMENTAL LUBRICATION/ CONFERENCE/ ENVIRONMENT/ EVAPORATION/ FRICTION/ LUBRICATION/ MOLYBDENUM/ SPACE/ SULFIDE/ TEFLON

67A14995 ISSUE 4 PAGE 704 CATEGORY 31 66/00/00 25 PAGES
UNCLASSIFIED DOCUMENT

REVIEW OF THE INFLUENCE OF SPACE ENVIRONMENT UPON VEHICLE COMPONENTS.

(SPACECRAFT ENVIRONMENTAL EFFECTS COVERING OUTGASSING IN HIGH VACUUM, DETERIORATION OF MATERIALS BY EVAPORATION, LUBRICATION AND CHANGES IN MECHANICAL AND ELECTRICAL PROPERTIES OF PLASTICS)

A/SMELT, R. (AA/LOCKHEED AIRCRAFT CORP., LOCKHEED MISSILES AND SPACE CO., SPACE PROGRAMS DIV., SUNNYVALE, CALIF./.)

IN- THE FLUID DYNAMIC ASPECTS OF SPACE FLIGHT, PROCEEDINGS OF THE NATO-AGARD SPECIALISTS' MEETING, MARSEILLE, FRANCE, APR. 20-24, 1964. VOLUME 1. <A67-14987 04-12< MEETING SPONSORED BY THE FLUID DYNAMICS PANEL OF AGARD. NEW YORK, GORDON AND BREACH, SCIENCE PUBLISHERS, INC. /AGARDOGRAPH 87. VOLUME 1/, 1966, P. 141-165. 18 REFS.

/*HIGH VACUUM/*LUBRICATION/*RADIATION EFFECT/*SPACE ENVIRONMENT/*SPACECRAFT CONSTRUCTION MATERIAL/*VACUUM EFFECT/ AGENA ROCKET/ BEARING/ CELL/ CONFERENCE/ DAMAGE/ DEGASSING/ DETERIORATION/ ELECTRIC/ ENVIRONMENT/ FLARE/ HAZARD/ MATERIAL/ MECHANICAL/ METECRITE/ RADIATION/ SOLAR/ SPACE/ SURFACE

67A12453* ISSUE 2 PAGE 250 CATEGORY 15 66/1C/00 7 PAGES
UNCLASSIFIED DOCUMENT

LUBRICANTS AND MECHANICAL COMPONENTS OF LUBRICATION SYSTEMS FOR
SPACE ENVIRONMENT.

(SYSTEM DESIGN AND LUBRICATING MATERIALS FOR SPACECRAFT LUBRICATION)
A/BUCKLEY, C. H.; B/JOHNSON, R. L. (AB/NASA, LEWIS RESEARCH
CENTER, LUBRICATION BRANCH, CLEVELAND, OHIO/ AA/NASA, LEWIS RESEARCH
CENTER, SPACE ENVIRONMENT LUBRICATION SECTION, CLEVELAND, OHIO/.)

/AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 19TH,
CHICAGO, ILL., MAY 26-28, 1964, PAPER./ LUBRICATION ENGINEERING, VOL.
22, OCT. 1966, P. 408-414. 45 REFS. <FOR ABSTRACT SEE ISSUE 12, PAGE
912, ACCESSION NO. A64-17992<

/*LUBRICANT/*SPACE ENVIRONMENTAL LUBRICATION/ CONFERENCE/ DESIGN/
ENVIRONMENT/ LUBRICATION/ MATERIAL/ SPACE/ SPACE VEHICLE/ SYSTEM

66A42573 ISSUE 24 PAGE 4261 CATEGORY 28 66/08/00 5 PAGES
UNCLASSIFIED DOCUMENT

THE INFLUENCE OF SPACE ENVIRONMENT ON THE DESIGN AND DEVELOPMENT OF
ROCKET ENGINES FOR SPACE APPLICATIONS.

(ENVIRONMENT INFLUENCE ON DESIGN AND DEVELOPMENT OF BLACK KNIGHT
ROCKET ENGINE)

A/SUNLEY, H. L. G. (AA/BRISTOL SIDDELEY ENGINES, LTD., LONDON,
ENGLAND/.)

/SOCIETY OF ENVIRONMENTAL ENGINEERS, SYMPOSIUM ON ENVIRONMENTAL
ENGINEERING AND ITS ROLE IN SOCIETY, IMPERIAL COLL. OF SCIENCE AND
TECHNOLOGY, LONDON, ENGLAND, APR. 19-21, 1966. VOLUME 5 - SPACE
TECHNOLOGY./ JOURNAL OF ENVIRONMENTAL SCIENCES, VOL. 9, AUG. 1966, P.
13-17. <FOR ABSTRACT SEE ISSUE 18, PAGE 3163, ACCESSION NO. A66-34211<

/*BLACK KNIGHT ROCKET/*ENVIRONMENTAL TESTING/*ROCKET ENGINE
DESIGN/*SPACE ENVIRONMENT/ ACCELERATION/ ALTITUDE/ CONFERENCE/
ELECTRIC/ ENVIRONMENT/ LAUNCHER/ LUBRICATION/ ROCKET ENGINE/ SATELLITE/
SPACE/ SYSTEM/ TESTING/ TURBOPUMP/ VIBRATION

66A35833# ISSUE 19 PAGE 3363 CATEGORY 15 66/00/00 9 PAGES
UNCLASSIFIED DOCUMENT

SOME EXPERIMENTS ON DRY LUBRICATION.

(TRANSFER OF DRY LUBRICANT PARTICLES ON TO CONTACT SURFACES DURING
SERVICE BY USING SEPARATE RUBBING ELEMENTS, FOR EQUIPMENT OPERATION IN
SPACE ENVIRONMENT)

A/ONARAN, K. (AA/ISTANBUL, TECHNICAL U., ENGINEERING MATERIALS
DEPT., ISTANBUL, TURKEY/.)

ISTANBUL TEKNİK UNIVERSİTESİ BÜLTENİ, VOL. 18, NO. 2, 1966, P.
34-42. 11 REFS.

/*SOLID LUBRICANT/*SPACE ENVIRONMENTAL LUBRICATION/ DRY/ ELEMENT/
ENVIRONMENT/ EQUIPMENT/ FRICTION/ LUBRICANT/ LUBRICATION/ MATERIAL/
PARTICLE/ RELIABILITY/ SOLID/ SPACE/ SURFACE/ TRANSFER

66A34211# ISSUE 18 PAGE 3163 CATEGORY 28 66/00/00 16 PAGES
UNCLASSIFIED DOCUMENT

THE INFLUENCE OF ENVIRONMENT ON THE DESIGN AND DEVELOPMENT OF ROCKET
ENGINES FOR SPACE APPLICATIONS.

(ENVIRONMENT INFLUENCE ON DESIGN AND DEVELOPMENT OF BLACK KNIGHT
ROCKET ENGINE)

A/SUNLEY, H. L. G. (AA/BRISTOL SIDDELEY ENGINES, LTD., ROCKET
DEPT., COVENTRY, WARWICKS., ENGLAND/.)

IN- SOCIETY OF ENVIRONMENTAL ENGINEERS, SYMPOSIUM ON
ENVIRONMENTAL ENGINEERING AND ITS ROLE IN SOCIETY, IMPERIAL COLL. OF
SCIENCE AND TECHNOLOGY, LONDON, ENGLAND, APR. 19-21, 1966. VOLUME 5 -
SPACE TECHNOLOGY <A66-34208 1E-3G< 1966. 16 P.

/*BLACK KNIGHT ROCKET/*ENVIRONMENTAL TESTING/*ROCKET ENGINE
DESIGN/*SPACE ENVIRONMENT/ ACCELERATION/ ALTITUDE/ CONFERENCE/
ELECTRIC/ ENVIRONMENT/ LAUNCHER/ LUBRICATION/ ROCKET ENGINE/ SATELLITE/
SPACE/ SYSTEM/ TESTING/ TURBOPUMP/ VIBRATION

66A30409 ISSUE 16 PAGE 2711 CATEGORY 15 ASLE PAPER 66AM 3C2
66/05/00 16 PAGES UNCLASSIFIED DOCUMENT

ADVANCED AEROSPACE GREASES.

(GREASE LUBRICANTS FOR AEROSPACE APPLICATION, DETERMINING PHYSICAL
PROPERTIES AND TESTING THEM AT 400 DEGREES F AND UNDER HIGH VACUUM)

A/BUNTING, K. R.; B/CHRISTIAN, J. B. (AB/USAF, WRIGHT-PATTERSON
AFB, OHIO/ AA/AMERICAN OIL CO., CHICAGO, ILL./.) MEMBERS, \$0.60,
NONMEMBERS, \$1.20.

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 21ST,
PITTSBURGH, PA., MAY 2-5, 1966, PAPER 66AM 3C2. 16 P.

/*GREASE/*LUBRICANT/*PHYSICAL PROPERTY/*SPACE ENVIRONMENTAL
LUBRICATION/ AEROSPACE/ BEARING/ CONFERENCE/ ENVIRONMENT/ LOAD/
LUBRICATION/ PHYSICAL/ PRESSURE/ PROPERTY/ SPACE/ SPEED/ TECHNOLOGY/
TEMPERATURE/ TIME/ VACUUM

66A25770 ISSUE 13 PAGE 2086 CATEGORY 15 65/11/00 4 PAGES
UNCLASSIFIED DOCUMENT

LUBRICANTS FOR THE SPACE ENVIRONMENT.

(LUBRICANTS FOR SPACE ENVIRONMENT, NOTING EXPERIENCES WITH
SATELLITES AND APPLICATION TO BALL BEARINGS AND ELECTRICAL CONTACTS)

A/CLAUSS, F. J. (AA/LOCKHEED AIRCRAFT CORP., LOCKHEED MISSILES AND
SPACE CO., PALO ALTO, CALIF./.)

/AMERICAN SOCIETY FOR METALS, WESTERN METAL CONGRESS, LOS
ANGELES, CALIF., FEB. 22-26, 1965, PAPER./ METALS ENGINEERING
QUARTERLY, VOL. 5, NOV. 1965, P. 58-61.

/*BALL BEARING/*ELECTRIC CONTACT/*SPACE ENVIRONMENTAL
LUBRICATION/*SPACE MAINTENANCE/ BALL/ BEARING/ CONFERENCE/ CONTACT/
ELECTRIC/ ENVIRONMENT/ LUBRICATION/ MAINTENANCE/ SPACE

66A18284* ISSUE 7 PAGE 1039 CATEGORY 15 ASLE PREPRINT 65AM 5C5
65/10/00 16 PAGES UNCLASSIFIED DOCUMENT

FLUORIDE SOLID LUBRICANTS FOR EXTREME TEMPERATURES AND CORROSIVE ENVIRONMENTS.

(CHEMICAL AND THERMAL STABILITY OF FLUORIDE SOLID LUBRICANTS MEASURED AND TESTED FOR AEROSPACE ENVIRONMENT)

A/ALLEN, G. P.; B/SLINEY, H. E.; C/STROM, T. N. (AA/NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO/.)

/AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 20TH, DETROIT, MICH., MAY 4-7, 1965, PREPRINT 65AM 5C5./ ASLE TRANSACTIONS, VOL. 8, OCT. 1965, P. 307-318, DISCUSSION, C. D. STUBER /USAF, WRIGHT PATTERSON AIR DEVELOPMENT CENTER, DAYTON, OHIO/, J. A. NELSON /ILLINOIS, U., DEPT. OF CERAMIC ENGINEERING, URBANA, ILL./, AND E. S. BOBER /WESTINGHOUSE ELECTRIC CORP., RESEARCH LABS., PITTSBURGH, PA./, P. 319-321, AUTHORS' CLOSURE, P. 321, 322. 11 REFS. <FOR ABSTRACT SEE ISSUE 14, PAGE 2034, ACCESSION NO. A65-24249<

/*CALCIUM FLUORIDE/*SOLID LUBRICANT/*SPACE ENVIRONMENTAL LUBRICATION/ AIR/ CALCIUM/ CHEMICAL/ COATING/ CONFERENCE/ ENVIRONMENT/ FLUORIDE/ FRICTION/ HYDROGEN/ LUBRICANT/ LUBRICATION/ METAL/ SODIUM/ SOLID/ SPACE/ STABILITY/ THERMAL/ WEAR

66A13398 ISSUE 4 PAGE 524 CATEGORY 15 ASLE PREPRINT 65AM 5C4
AF 29/601/-6213 65/11/00 11 PAGES UNCLASSIFIED DOCUMENT

THE EFFECTS OF REACTOR RADIATION ON THREE HIGH- TEMPERATURE SOLID-FILM LUBRICANTS.

(GAMMA RAY AND NEUTRON IRRADIATION AND TEMPERATURE EFFECTS ON WEAR-LIFE OF SOLID LUBRICANTS FOR AEROSPACE APPLICATION)

A/MCDANIEL, R. H. (AA/GENERAL DYNAMICS CORP., GENERAL DYNAMICS/FORT WORTH, NUCLEAR AEROSPACE RESEARCH FACILITY, FORT WORTH, TEX./.)

/AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 20TH, DETROIT, MICH., MAY 4-7, 1965, PREPRINT 65AM 5C4./ LUBRICATION ENGINEERING, VOL. 21, NOV. 1965, P. 463-471, DISCUSSION, H. E. SLINNEY /NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO/, MEL LAVIK /MIDWEST RESEARCH INST., KANSAS CITY, MO./, P. 472, AUTHOR'S CLOSURE, P. 473. 11 REFS. <FOR ABSTRACT SEE ISSUE 14, PAGE 2034, ACCESSION NO. A65-24248<

/*NUCLEAR RADIATION/*RADIATION EFFECT/*SOLID LUBRICANT/*SPACE ENVIRONMENTAL LUBRICATION/*TEMPERATURE EFFECT/ CALCIUM/ CONFERENCE/ EFFECT/ ENVIRONMENT/ GAMMA/ LEAD/ LIFE/ LUBRICANT/ LUBRICATION/ NICKEL/ NUCLEAR/ RADIATION/ REACTOR/ SILICATE/ SOLID/ SPACE/ STEEL/ TEMPERATURE/ WEAR

65A33076 ISSUE 21 PAGE 3128 CATEGORY 17 65/00/00 16 PAGES
UNCLASSIFIED DOCUMENT

MATERIAL REQUIREMENTS FOR SPACE APPLICATIONS.

(MATERIAL BEHAVIOR, PROTECTION AND SELECTION FOR VARIOUS SPACE ENVIRONMENTAL CONDITIONS)

A/HAUSNER, F. H.; B/WOLKOWITZ, W. (AB/BROOKLYN, POLYTECHNIC INST., BROOKLYN, N.Y., GRUMMAN AIRCRAFT ENGINEERING CORP., BETHPAGE, N.Y./.)

IN- METALS FOR THE SPACE AGE, PLANSEE SEMINAR DE RE METALLICA, 5TH, REUTTE, AUSTRIA, JUN. 22- 26, 1964, PROCEEDINGS. <A65-33073 21-17< EDITED BY F. BENESOVSKY. REUTTE, AUSTRIA, METALLWERK PLANSEE AG, 1965, P. 61-76.

/*PROTECTIVE COATING/*REFRACTORY MATERIAL/*SPACE ENVIRONMENT/*STRUCTURAL MATERIAL/ AEROSPACE/ COATING/ CONFERENCE/ ENVIRONMENT/ FILM/ FRICTION/ HIGH TEMPERATURE/ HIGH VACUUM/ LIFE/ LUBRICATION/ MATERIAL/ PROTECTION/ REENTRY/ REFRACTORY/ SPACE/ STRUCTURAL/ TECHNOLOGY

65A31144 ISSUE 19 PAGE 2747 CATEGORY 3 65/06/00 8 PAGES
UNCLASSIFIED DOCUMENT

ROTATING MACHINES FOR EXTREME ENVIRONMENTS.

(ROTATING MACHINES IN EXTREME ENVIRONMENT DISCUSSING CONDUCTORS, MAGNETIC MATERIALS, INSULATIONS, BEARINGS AND PERFORMANCE)

A/IRANI, D.; B/SMITH, C. S. (AB/GARRETT CORP., AIRESEARCH MFG. CO., LOS ANGELES, CALIF./.)

/1965 AEROSPACE TECHNICAL CONFERENCE AND EXHIBIT, HOUSTON, TEX., JUN. 21-24, 1965, PAPER./ IEEE TRANSACTIONS ON AEROSPACE, VOL. AS-3, JUN. 1965, SUPPLEMENT, P. 620-627. 25 REFS.

/*BEARING/*LUBRICATION SYSTEM/*RADIATION EFFECT/*ROTATING MACHINE/*SPACE ENVIRONMENT/*TEMPERATURE EFFECT/*VACUUM EFFECT/ CERAMICS/ CONFERENCE/ DESIGN/ EFFECT/ ELECTRIC/ ENVIRONMENT/ HEAT TRANSFER/ INSULATION/ LUBRICATION/ MACHINE/ RADIATION/ ROTATION/ SPACE/ SYSTEM/ TEMPERATURE/ VACUUM

65A24249* ISSUE 14 PAGE 2034 CATEGORY 15 ASLE PREPRINT 65AM 5C5
65/05/00 31 PAGES UNCLASSIFIED DOCUMENT

FLUORIDE SOLID LUBRICANTS FOR EXTREME TEMPERATURES AND CORROSIVE ENVIRONMENTS.

(CHEMICAL AND THERMAL STABILITY OF FLUORIDE SOLID LUBRICANTS MEASURED AND TESTED FOR AEROSPACE ENVIRONMENT)

A/ALLEN, G. P.; B/SLINEY, F. E.; C/STROM, T. N. (AA/NASA, LEWIS RESEARCH CENTER, CLEVELAND, OHIO/.) MEMBERS, \$0.60, NONMEMBERS, \$1.20.

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 20TH, DETROIT, MICH., MAY 4-7, 1965, PREPRINT 65AM 5C5. 31 P. 11 REFS.

/*CALCIUM FLUORIDE/*SOLID LUBRICANT/*SPACE ENVIRONMENTAL LUBRICATION/ AIR/ CALCIUM/ CHEMICAL/ COATING/ CONFERENCE/ ENVIRONMENT/ FLUORIDE/ FRICTION/ HYDROGEN/ LUBRICANT/ LUBRICATION/ METAL/ SODIUM/ SOLID/ SPACE/ STABILITY/ THERMAL/ WEAR

65A24248 ISSUE 14 PAGE 2034 CATEGORY 15 ASLE PREPRINT 65AM 5C4
AF 29/601/-6213 65/05/00 30 PAGES UNCLASSIFIED DOCUMENT
THE EFFECTS OF REACTOR RADIATION ON THREE HIGH- TEMPERATURE
SOLID-FILM LUBRICANTS.

(GAMMA RAY AND NEUTRON IRRADIATION AND TEMPERATURE EFFECTS ON
WEAR-LIFE OF SOLID LUBRICANTS FOR AEROSPACE APPLICATION)

A/MCDANIFL, R. H. (AA/GENERAL DYNAMICS CORP., NUCLEAR AEROSPACE
RESEARCH FACILITY, FORT WORTH, TEX./.) MEMBERS, \$0.60, NONMEMBERS,
\$1.20.

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 20TH,
DETROIT, MICH., MAY 4-7, 1965, PREPRINT 65AM 5C4. 30 P. 11 REFS.

/*NUCLEAR RADIATION/*RADIATION EFFECT/*SOLID LUBRICANT/*SPACE
ENVIRONMENTAL LUBRICATION/*TEMPERATURE EFFECT/ CALCIUM/ CONFERENCE/
EFFECT/ ENVIRONMENT/ GAMMA/ LEAD/ LIFE/ LUBRICANT/ LUBRICATION/ NICKEL/
NUCLEAR/ RADIATION/ REACTOR/ SILICATE/ SOLID/ SPACE/ STEEL/
TEMPERATURE/ WEAR

65A24245 ISSUE 14 PAGE 2033 CATEGORY 15 ASLE PREPRINT 65AM 5C3
65/05/00 26 PAGES UNCLASSIFIED DOCUMENT

NEW SOLID LUBRICANTS - PREPARATION, PROPERTIES AND POTENTIALS.

(PHYSICAL AND CHEMICAL PROPERTIES OF DICHALCOGENIDES OF GROUP VB AND
VIB METALS FOR USE AS AEROSPACE SOLID LUBRICANTS, PRIMARILY FRICTION
AND ANTIWEAR CHARACTERISTICS)

A/BOES, D. J. (AA/WESTINGHOUSE ELECTRIC CORP., RESEARCH LAB.,
PITTSBURGH, PA./.) MEMBERS, \$0.60, NONMEMBERS, \$1.20.

AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 20TH,
DETROIT, MICH., MAY 4-7, 1965, PREPRINT 65AM 5C3. 26 P. 11 REFS.

/*FRICTION COEFFICIENT/*REFRACTORY METAL/*SOLID LUBRICANT/*SPACE
ENVIRONMENTAL LUBRICATION/ CHALCOGENIDE/ CHEMICAL/ COEFFICIENT/
CONFERENCE/ CRYSTAL/ DISULFIDE/ ENVIRONMENT/ FRICTION/ GRAPHITE/
LUBRICANT/ METAL/ MOLYBDENUM/ NIOBIUM/ PERIODIC GROUP VB/ PERIODIC
GROUP VIB/ PHYSICAL/ PROPERTY/ REFRACTORY/ SELENIDE/ SOLID/ SPACE/
STRUCTURE/ TANTALUM/ TELLURIDE/ TUNGSTEN/ WEAR

65A23451# ISSUE 13 PAGE 1876 CATEGORY 15 ASME PAPER 65-GTP-14
65/02/00 20 PAGES UNCLASSIFIED DOCUMENT

LOW-LEAKAGE DYNAMIC SEAL TO SPACE.

(HIGH SPEED LOW LEAKAGE SEAL-TO-SPACE USING SLINGER, VISCO AND
MOLECULAR PUMP SEALS IN MERCURY RANKINE CYCLE)

A/HODGSON, J. N.; B/LESSLEY, R. L. (AA/AEROJET-GENERAL CORP.,
SNAP-8 DIV., AZUSA, CALIF./.) MEMBERS, \$0.50, NONMEMBERS, \$1.00.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS, GAS TURBINE CONFERENCE
AND PRODUCTS SHOW, WASHINGTON, D.C., FEB. 28-MAR. 4, 1965, PAPER
65-GTP-14. 20 P. 22 REFS.

/*LIQUID-VAPOR INTERFACE/*MERCURY VAPOR/*MOLECULAR PUMP/*PUMP
SEAL/*RANKINE CYCLE/*SPACE ENVIRONMENT/ BEARING/ CONFERENCE/
ENVIRONMENT/ INTERFACE/ LEAKAGE/ LIQUID/ LUBRICATION/ MERCURY/
MOLECULAR/ OIL/ POWER/ PUMP/ SEAL/ SPACE/ SPEED/ SYSTEM/ TEST/ TURBINE/
VACUUM/ VAPOR

65A22751 ISSUE 12 PAGE 1720 CATEGORY 18 65/CO/00 37 PAGES
UNCLASSIFIED DOCUMENT

SELECTION OF MATERIALS FOR OPTIMUM PERFORMANCE.

(ENVIRONMENTAL CONSIDERATIONS IN SPACECRAFT CONSTRUCTION MATERIAL
SELECTION FOR ADHESIVES, SEALS AND THERMAL, OPTICAL, LUBRICATED AND
ELECTRONIC SYSTEMS)

IN- SPACE MATERIALS HANDBOOK. EDITED BY C. G. GOETZEL, J. B.
RITTENHOUSE, AND J. B. SINGLETARY. READING, MASS., ADDISON- WESLEY
PUBLISHING CO., INC., 1965, P. 519-555. 10 REFS.

/*SPACE ENVIRONMENT/*SPACECRAFT CONSTRUCTION MATERIAL/ ADHESIVE/
CONSTRUCTION/ CONTROL/ ELECTRONICS/ ENVIRONMENT/ INORGANIC/
LUBRICATION/ MATERIAL/ OPTICS/ OPTIMUM/ POLYMER/ SEALING/ SELECTION/
SPACE/ SPACECRAFT/ SURFACE/ SYSTEM/ THERMAL

65A22733 ISSUE 12 PAGE 1718 CATEGORY 17 AF 36/657/-10107
65/00/00 624 PAGES UNCLASSIFIED DOCUMENT

SPACE MATERIALS HANDBOOK.

(HANDBOOK ON SPACE MATERIALS NOTING SPACE ENVIRONMENTS) \$15.

EDITED BY C. G. GOETZEL, J. B. RITTENHOUSE, AND J. B. SINGLETARY
/LOCKHEED AIRCRAFT CORP., LOCKHEED MISSILES AND SPACE CO., SUNNYVALE,
CALIF./ . READING, MASS., ADDISON-WESLEY PUBLISHING CO., INC., 1965.
624 P.

/*MATERIALS SCIENCE/*SPACE ENVIRONMENT/*SPACECRAFT CONSTRUCTION
MATERIAL/ ADHESIVE/ CERAMIC/ CONSTRUCTION/ ELECTRONIC/ ENVIRONMENT/
HANDBOOK/ LUBRICATION/ MATERIAL/ OPTICAL/ RADIATION/ SCIENCE/ SPACE/
SPACECRAFT/ THERMAL

65A13607 ISSUE 4 PAGE 470 CATEGORY 15 AF 33/657/-7395
64/12/00 7 PAGES UNCLASSIFIED DOCUMENT

DEVELOPMENT AND EVALUATION OF LUBRICANT COMPOSITE MATERIALS.

(MOLYBDENUM DISULFIDE HOT PRESSING WITH METAL BINDERS TO FORM
LUBRICANT COMPOSITE MATERIALS AND BEARING TEST DATA)

A/CAMPBELL, M. E.; B/VAN WYK, J. W. (AA/MIDWEST RESEARCH INST.,
KANSAS CITY, MO./ AB/BOEING CO., SEATTLE, WASH./.)

/AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING, 19TH,
CHICAGO, ILL., MAY 26-28, 1964./ LUBRICATION ENGINEERING, VOL. 20,
DEC. 1964, P. 463-469.

/*COMPOSITE MATERIAL/*HOT PRESSING/*MOLYBDENUM SULFIDE/*SOLID
LUBRICANT/*SPACE ENVIRONMENTAL LUBRICATION/ BEARING/ DISULFIDE/
ENVIRONMENT/ FRICTION/ HIGH TEMPERATURE/ IRON/ LUBRICANT/ MATRIX/
METALLURGY/ MOLYBDENUM/ PLATINUM/ POWDER/ SOLID/ SPACE/ TEST

64A24312 ISSUE 20 CATEGORY 17 DA-36-039-SC-89207 64/07/00 13
PAGES UNCLASSIFIED DOCUMENT

LONG-DURATION LUBRICATION STUDIES IN SIMULATED SPACE VACUUM.

(LONG DURATION TESTING OF SLIDER AND BALL BEARING LUBRICANTS IN
SIMULATED SPACE ENVIRONMENT)

A/BROWN, R. D.; B/BURTON, R. A.; C/KU, P. M. (AC/SOUTHWEST
RESEARCH INST., SAN ANTONIO, TEX./.)

ASLE TRANSACTIONS, VOL. 7, JUL. 1964, P. 236- 248. 13 REFS.
RESEARCH SPONSORED BY THE SOUTHWEST RESEARCH INST.

/*BALL BEARING/*LUBRICATION TESTING MACHINE/*SPACE ENVIRONMENTAL
LUBRICATION/ BEARING/ DURATION/ ENVIRONMENT/ LONG/ SLIDING/ TESTING/
VACUUM

64A24311 ISSUE 20 CATEGORY 17 AF 40/600/-915

64/07/00 9 PAGES UNCLASSIFIED DOCUMENT

SOLID LUBRICATION OF GEARS AND BEARINGS IN A SPACE ENVIRONMENT.

(COMPOSITE SOLID LUBRICANTS FOR SPACE LUBRICATION OF LOADED GEARS
AND BEARINGS, INCLUDING TECHNIQUES AND TEST EQUIPMENT)

A/BOWEN, P. H. (AA/WESTINGHOUSE ELECTRIC CORP., RESEARCH LABS.,
PITTSBURGH, PA./.)

/AMERICAN SOCIETY OF LUBRICATION ENGINEERS, ANNUAL MEETING,
CHICAGO, ILL., MAY 1964./ ASLE TRANSACTIONS, VOL. 7, JUL. 1964, P.
227-235. 6 REFS.

/*LUBRICATION TESTING MACHINE/*SOLID LUBRICANT/*SPACE ENVIRONMENTAL
LUBRICATION/ BALL/ BEARING/ COMPOSITE/ ENVIRONMENT/ EQUIPMENT/ GEAR/
LOAD/ MACHINE/ MATERIAL/ TECHNIQUE/ TESTING/ VACUUM

64A22749* ISSUE 19 CATEGORY 17 NAS5-814 64/07/00 5 PAGES
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FRICTION VARIATION OF PTFE AND MCS SUB 2 DURING THERMAL VACUUM
EXPOSURE.

(FRICTION COEFFICIENTS OF POLYTETRAFLUOROETHYLENE AND MOLYBDENUM
SULFIDE-GRAPHITE-SODIUM SILICATE COATINGS FOR SPACE LUBRICATION)

A/CRAIG, W. D., JR. (AA/GRUMMAN AIRCRAFT ENGINEERING CORP.,
MECHANICAL SYSTEMS SECTION, BETHPAGE, N.Y./.)

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LUBRICANT/*TEFLON/ COATING/ FILM/ GRAPHITE/ LUBRICATION/ SILICATE/
SPACE ENVIRONMENTAL LUBRICATION/ THIN/ VEHICLE

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GREASE LUBRICANTS AND THEIR POTENTIAL IN AEROSPACE APPLICATIONS.
(GREASE LUBRICANTS FOR AEROSPACE VEHICLES AND THEIR SUPPORT
EQUIPMENT)

A/SCHWENKER, H. S. (SCHWENKER, H. /USAF, SYSTEMS COMMAND,
RESEARCH AND TECHNOLOGY DIV., AF MATERIALS LAB., FLUID AND LUBRICANT
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64A19124* ISSUE 14 CATEGORY 17 64/01/00 9 PAGES UNCLASSIFIED
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LUBRICANT BEHAVIOR IN HIGH VACUUM.

(HIGH VACUUM EFFECTS ON DRY FRICTION COEFFICIENT, LUBRICATED
FRICTION COEFFICIENT AND LOAD CARRYING CAPACITY OF LUBRICANTS)

A/FOSTER, P. G.; B/REICHENBACH, G. S.; C/SHAW, R., JR.
(AA/MASSACHUSETTS INST. OF TECH., SURFACE LAB., DEPT. OF MECHANICAL
ENGINEERING, CAMBRIDGE, MASS./.)

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VACUUM/ LOAD/ LUBRICATION/ PIN/ PRESSURE/ TEST

64A17992# ISSUE 12 CATEGORY 17 00/00/00 UNCLASSIFIED DOCUMENT
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SPACE ENVIRONMENT.

(SYSTEM DESIGN AND LUBRICATING MATERIALS FOR SPACECRAFT LUBRICATION)

A/BUCKLEY, D. H.; B/JOHNSON, R. L. (AA/NASA, LEWIS RESEARCH
CENTER, CLEVELAND, OHIO/.)

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LUBRICATION/ MATERIAL/ SPACE/ SPACE VEHICLE/ SYSTEM

63A25801 ISSUE 24 CATEGORY 17 63/00/00 48 PAGES UNCLASSIFIED
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(ENVIRONMENTAL CONDITIONS AND OPERATING CHARACTERISTICS OF
SPACECRAFT LUBRICATION, NOTING SILICONE OILS AND GREASES)

A/CLAUSS, F. J.

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GREASE/ HIGH VACUUM/ LUBRICANT/ LUBRICATION/ OIL/ SILICONE/ SPACE/
SPACECRAFT

63A23404 ISSUE 22 CATEGORY 18 63/09/00 3 PAGES UNCLASSIFIED
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(SUBLIMATION AND OUT-GASSING ARE MAIN FACTORS CONSIDERED IN
SELECTION OF SUITABLE METALS, CERAMICS AND PLASTICS FOR SPACE
APPLICATIONS)

A/JAHNKE, C. E.

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VACUUM/ VAPOR

63A20497* ISSUE 19 CATEGORY 17 NAS7-100 63/08/00 2 PAGES
UNCLASSIFIED DOCUMENT

(COEFFICIENT OF FRICTION UNDER SPACE ENVIRONMENT IN THE RANGER I
SPACECRAFT)

A/JAFFE, L. D.; B/MARTENS, H. E.; C/NAGLER, R. G.; D/RITTENHOUSE,
J. B.

RESULTS OF RANGER I FLIGHT FRICTION EXPERIMENT. J. B.
RITTENHOUSE, L. D. JAFFE, R. G. NAGLER, AND H. E. MARTENS /CALIFORNIA
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MEASUREMENT/ METAL/ SPACE

63A12681* ISSUE 6 CATEGORY 17 ARS PAPER 62-2711 62/11/00 11
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(HIGH-SPEED OPERATION OF MINIATURE BALL BEARINGS WITH METALLIC FILM
LUBRICATION AND BEST-GEAR MATERIAL COMBINATIONS FOR VACUUM-SPACE
OPERATION)

A/EVANS, H. E.; B/FEDERLINE, M. F.; C/FLATLEY, T. W.

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(STUDY OF PROVIDING LUBRICATION FOR REDUCING FRICTION AND WEAR OF
RUBBING OR SLIDING SURFACES OF VARIOUS SPACECRAFT MECHANISMS)

A/CLAUSS, F. J.; B/COOKE, F. B.; C/DRAKE, S. P.; D/CHARA, C. F.

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C. F. O'HARA, S. P. DRAKE, AND F. B. COOKE /LOCKHEED AIRCRAFT CORP.,
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LAMINAR/ LUBRICATION/ MATERIAL/ MECHANISM/ MOLYBDENUM/ OIL/ PLASTICITY/
SLIDE/ SOLID/ SPACE/ SPACECRAFT/ SURFACE

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OF SLIP RING IN A SPACE ENVIRONMENT FINAL REPORT

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LIFE (DURABILITY)/ LUBRICATION/ OSCILLOGRAPHS/ RELIABILITY/ SLIDING
FRICTION/ VACUUM/ VELOCITY

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A/BENTALL, R. H.

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TESTS/ SPACE ENVIRONMENT SIMULATION

73N32385# ISSUE 23 PAGE 2799 CATEGORY 15 AD-765300 AD-702550
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71N33011# ISSUE 20 CATEGORY 15 BMBW-FB-W-71-11 71/03/00 62
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PROJECT HELIOS - DEVELOPMENT WORK ON AN OIL VAPOR LUBRICATED ANTENNA
BEARING SYSTEM

(VAPORIZED OIL BALL BEARINGS FOR SATELLITE ANTENNAS)

A/BAUMGARTNER, G.; B/GRANZOW, M.; C/HOSTENKAMP, R.; D/WOHNHAAS,
H.

DORNIER-WERKE G.M.B.H., FRIEDRICHSHAFEN (WEST GERMANY).

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SPACECRAFT LUBRICATION/ STORAGE TANKS/ TORQUE

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ENVIRONMENTAL CONDITIONS
(SPACE ENVIRONMENT SIMULATION TESTING OF ANTENNA SPIN REDUCTION
BEARING AND MECHANICAL DRIVES FOR HELIOS SOLAR PROBE)
A/BIRKHOFF, E.; B/GRANZOW, M.; C/GROSSNER, P.; D/HAUSER, G.;
E/TREFFON, H. R.

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SIMULATION/*SPACECRAFT ANTENNAS/*SPIN REDUCTION/ BALL BEARINGS/
ENCAPSULATING/ LUBRICATING OILS/ PARABOLIC ANTENNAS/ ROTATING BODIES/
SPACECRAFT LUBRICATION

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70/11/00 79 PAGES UNCLASSIFIED DOCUMENT

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(COMMUNICATION SATELLITE DIRECTIONAL ANTENNA SPIN REDUCTION
MECHANISM)

DORNIER-WERKE G.M.B.H., FRIEDRICHSHAFEN (WEST GERMANY).
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ANTENNAS/*SPIN REDUCTION/ BALL BEARINGS/ ERROR ANALYSIS/ RETAINING/
SHAFTS (MACHINE ELEMENTS)/ SPACECRAFT LUBRICATION/ STEPS/ SYNCHRONOUS
MOTORS/ TEMPERATURE CONTROL/ TORQUE MOTORS

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SPACECRAFT LUBRICATION/ STRUCTURAL DESIGN/ TORQUE MOTORS

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(ACTUATORS AND BEARINGS FOR SATELLITE LOUVER THERMAL CONTROL ARRAY)
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(SELF LUBRICATING FLUORIDE-METAL COMPOSITE MATERIALS FOR OUTER SPACE
APPLICATIONS)
A/SLINEY, F. E. (AAINVENTOR /TC NASA/)
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
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MATERIALS

71N17865*# ISSUE 7 PAGE 976 CATEGORY 3 NASA-TM-X-2190 E-5957
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(MECHANICAL AND ELECTRICAL DESIGN OF ACTUATOR FOR SERT 2 SPACECRAFT)
A/HURST, E. B.; B/PROVENCHER, C. E., JR.; C/RENZ, D. D.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO. AVAIL. NTIS
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SPACECRAFT/ ENVIRONMENTAL TESTS/ PERFORMANCE TESTS/ SPACECRAFT
LUBRICATION

70N23759# ISSUE 10 PAGE 1934 CATEGORY 31 RAE-TR-69055
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(OIL VAPOR CONTAMINATION EFFECTS OF SATELLITE OPTICAL SURFACES
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A/WILLIAMS, T. N.
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SIMULATION/*SPACECRAFT CONTAMINATION/*ULTRAVIOLET RADIATION/ SOLAR
CELLS/ SURFACE REACTIONS/ VAPOR DEPOSITION

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(PROPERTIES ANALYSIS OF LUBRICATING MATERIALS USED WITH SPACECRAFT
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LOCKHEED MISSILES AND SPACE CO., PALO ALTO, CALIF. (RESEARCH LAB.)
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SURFACE PROPERTIES/ TEMPERATURE EFFECTS

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EFFECT OF OXIDE FILM ON THE FRICTION OF 52100 STEEL IN VACUUM
(COEFFICIENT OF FRICTION DROP IN STEEL IN HIGH VACUUM)

A/TAKAGI, R.

DAYTON UNIV. RESEARCH INST., OHIO. AVAIL. NTIS

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FRICTION/ LUBRICANTS/ METAL PLATES/ OXIDES/ STRAIN GAGES

69N35503# ISSUE 20 PAGE 3806 CATEGORY 15 DLR-FB-69-21 DVL-858
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ON LUBRICATION PROPERTIES OF MOLYBDENUM DISULFIDE
POLYTETRAFLUOROETHYLENE DOUBLE LAYERS IN HIGH VACUUM
(MOLYBDENUM DISULFIDE AND POLYTETRAFLUOROETHYLENE DOUBLE LAYERS FOR
COATING SHAFT BEARING SURFACES)

A/HELLWIG, W.; B/SPENGLER, G.

DEUTSCHE VERSUCHSANSTALT FUER LUFT- UND RAUMFAHRT, MUNICH (WEST
GERMANY). (INSTITUT FUER FLUGTREIB- UND SCHMIERSTOFFE.) AVAIL. NTIS

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DISULFIDES/*POLYTETRAFLUOROETHYLENE/*SHAFTS (MACHINE ELEMENTS)/*THIN
FILMS/ ACTIVATION/ COEFFICIENT OF FRICTION/ HIGH VACUUM/ WEIGHT
MEASUREMENT

69N32381*# ISSUE 18 PAGE 3381 CATEGORY 15 NASA-TM-X-63606
X-723-69-226 69/05/00 73 PAGES UNCLASSIFIED DOCUMENT
DESIGN AND DEVELOPMENT OF A RADIAL LOAD BALL BEARING TEST SYSTEM AND
SOME PRELIMINARY RESULTS

(RADIAL LOAD BALL BEARING TEST SYSTEM WITH SPACE ENVIRONMENT
SIMULATION)

A/WALL, J. L.; B/WARD, B. W., JR.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. GODDARD SPACE FLIGHT
CENTER, GREENBELT, MD. AVAIL. NTIS

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TABLES (DATA)/ VACUUM SYSTEMS

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LUBRICATION AND BEARING PROBLEMS IN THE VACUUM OF SPACE
(SUMMARY INFORMATION ON LUBRICANTS AND LUBRICATION FOR USE IN
AEROSPACE ENVIRONMENTS, AND SIMULATION OF SPACE ENVIRONMENTS IN VACUUM
CHAMBER)

A/BISSON, E. E.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO. AVAIL. NTIS
PRESENTED AT THE SYMP. ON FRICTION DAYS SPONSORED BY GAMI,
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PERFORMANCE AND ANALYSIS OF SEALS FOR INERTED LUBRICATION SYSTEMS OF
TURBINE ENGINES

(PERFORMANCE EVALUATION OF NITROGEN INERTED LUBRICATION SYSTEM WITH
CONTACT SEALS FOR BALL BEARING OF TURBINE ENGINES)

A/JOHNSON, R. L.; B/LOOMIS, W. R.; C/LUDWIG, L. P.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO. AVAIL. NTIS
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TESTS/*TURBINE ENGINES/ INERTIA/ NITROGEN/ SEALS (STOPPERS)/ SPACE
ENVIRONMENT SIMULATION

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68/02/16 26 PAGES UNCLASSIFIED DOCUMENT

DRY FILM LUBRICATED BALL BEARINGS FOR GIMBALS OSCILLATING AT SMALL
ANGLES IN VACUUM

(OPERATIONAL PERFORMANCE TESTS ON DRY LUBRICATED BALL BEARINGS UNDER
SIMULATED GIMBAL OSCILLATION IN VACUUM)

A/DEMAREST, K. E.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. MARSHALL SPACE FLIGHT
CENTER, HUNTSVILLE, ALA. AVAIL. NTIS

/*BALL BEARINGS/*GIMBALS/*PERFORMANCE TESTS/*SLID
LUBRICANTS/*SPACE ENVIRONMENT SIMULATION/*STABLE OSCILLATIONS/
LUBRICANT TESTS/ OPERATIONS RESEARCH

68N19608** ISSUE 10 PAGE 1532 CATEGORY 15 NASA-TM-X-52208

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LUBRICATION AND BEARING PROBLEMS IN THE VACUUM OF SPACE
(EXPERIMENTS AND ENGINEERING MEASURES DEALING WITH LUBRICATION AND
BEARING PROBLEMS IN VACUUM OF SPACE)

A/BISSON, E. E.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO. AVAIL. NTIS

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/*ANTIFRICTION BEARINGS/*BEARING ALLOYS/*SPACECRAFT
LUBRICATION/*VACUUM EFFECTS/ AEROSPACE ENVIRONMENTS/ ATOM
CONCENTRATION/ BIBLIOGRAPHIES/ GRAVITATIONAL EFFECTS/ PRESSURE EFFECTS/
RADIATION EFFECTS/ SURFACE PROPERTIES/ TEMPERATURE EFFECTS

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SECOND AEROSPACE MECHANISMS SYMPOSIUM

(CONFERENCE PROCEEDINGS ON DESIGN AND PERFORMANCE OF AEROSPACE
MECHANISMS AND MECHANICAL COMPONENTS IN SPACE ENVIRONMENT)

A/HERZL, G. G. (AAED.)

JET PROPULSION LAB., CALIF. INST. OF TECH., PASADENA. AVAIL. NTIS

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/*CONFERENCE/*MECHANISM/*SPACE ENVIRONMENT/*SPACECRAFT COMPONENT/
AEROSPACE/ COMPONENT/ DEPLOYMENT/ DESIGN/ ENVIRONMENT/ LUBRICATION/
LUNAR SPACECRAFT/ PERFORMANCE/ RELEASE/ SATELLITE/ SIMULATOR/ SPACE/
SPACECRAFT/ SURVEYOR SPACECRAFT/ TEST

67N27281* ISSUE 14 PAGE 2557 CATEGORY 17 NASA-CR-72239 QPR-13
NAS3-2534 66/07/22 135 PAGES UNCLASSIFIED DOCUMENT

MATERIALS FOR POTASSIUM LUBRICATED JOURNAL BEARINGS QUARTERLY
PROGRESS REPORT, 22 APR. - 22 JUL. 1966

(FRICTION AND WEAR, AND COMPRESSION PROPERTIES OF CANDIDATE
LUBRICATED JOURNAL BEARINGS IN HIGH VACUUM AND IN LIQUID POTASSIUM)

A/FRANK, R. G. (AAED.)

GENERAL ELECTRIC CO., BURLINGTON, VT. (SPACE POWER AND PROPULSION
SECTION.)

22 JUL. 1966 135 P REFS

/*COMPRESSION/*FRICTION/*JOURNAL BEARING/*LIQUID POTASSIUM/ ALLOY/
BEARING/ CARBIDE/ HIGH VACUUM/ JOURNAL/ LIQUID/ LUBRICATION/ MATERIAL/
MOLYBDENUM/ POTASSIUM/ PROPERTY/ REFRACTORY/ STRESS/ TESTING/ TITANIUM/
WEAR

67N17831*# ISSUE 7 PAGE 1094 CATEGORY 15 NASA-TM-X-52271
67/00/00 42 PAGES UNCLASSIFIED DOCUMENT
LUBRICATION AND WEAR FUNDAMENTALS FOR HIGH VACUUM APPLICATIONS
(LUBRICATION, AND FRICTION AND WEAR CONCEPTS FOR HIGH VACUUM
APPLICATIONS)

A/BUCKLEY, D. H.; B/JOHNSON, R. L.
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH
CENTER, CLEVELAND, OHIO. AVAIL. NTIS
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ON LUBRICATION AND WEAR FUNDAMENTALS AND APPL. TO DESIGN, LONDON, 25-28
SEP. 1967, SPONSORED BY INST. OF MECH. ENGR.

/*HIGH VACUUM/*MATERIALS EROSION/*SLIDING FRICTION/*SPACE
ENVIRONMENTAL LUBRICATION/ ADHESION/ ALLOY/ BIBLIOGRAPHY/ CLEAN/
CLEAVAGE/ CONTAMINATION/ DEGRADATION/ ENVIRONMENT/ EROSION/ FILM/
FRICTION/ HARDWARE/ HIGH/ LUBRICATION/ MATERIAL/ METAL/ NEAR/ ORGANIC/
RESEARCH/ SLICING/ SOLID/ SPACE/ SPACECRAFT/ TECHNOLOGY/ VACUUM

67N15215*# ISSUE 5 PAGE 712 CATEGORY 15 NASA-CR-72041 QPR-12
NAS3-2534 66/04/22 192 PAGES UNCLASSIFIED DOCUMENT
MATERIALS FOR POTASSIUM LUBRICATED JOURNAL BEARINGS QUARTERLY
PROGRESS REPORT, 22 JAN. - 22 APR. 1966

(MATERIALS FOR POTASSIUM LUBRICATED JOURNAL BEARINGS)
A/FRANK, R. G. (AAED.)
GENERAL ELECTRIC CO., CINCINNATI, OHIO. (SPACE POWER AND
PROPULSION SECTION.) AVAIL. NTIS
22 APR. 1966 192 P REFS

/*JOURNAL BEARING/*LUBRICATION SYSTEM/*POTASSIUM/ ALLOY/ BALL/
BEARING/ CALIBRATION/ FLOW/ FRICTION/ HIGH VACUUM/ JOURNAL/
LUBRICATION/ MATERIAL/ METAL/ SYSTEM/ TEMPERATURE/ TEST/ VELOCITY/
WEAR

67N13850 ISSUE 4 PAGE 612 CATEGORY 30 66/00/00 25 PAGES
UNCLASSIFIED DOCUMENT

REVIEW OF THE INFLUENCE OF SPACE ENVIRONMENT UPON VEHICLE COMPONENTS
(ENVIRONMENTAL HAZARDS ENCOUNTERED IN SPACECRAFT FLIGHTS)
A/SMELT, R.

LOCKHEED MISSILES AND SPACE CO., SUNNYVALE, CALIF. (SPACE PROGRAMS
DIV.)

IN AGARD THE FLUID DYN. ASPECTS OF SPACE FLIGHT, VOL. 1 1966 P
141-165 REFS /SEE N67-13842 04-12/

/*FLIGHT HAZARD/*SPACECRAFT COMPONENT/*SPACECRAFT ENVIRONMENT/
AGFNA ROCKET/ BEARING/ COMPONENT/ CONFERENCE/ DYNAMICS/ ENVIRONMENT/
FLIGHT/ FLUID/ HAZARD/ LUBRICATION/ METEORITE/ OUTGASSING/ RADIATION/
SATELLITE/ SPACE FLIGHT/ SPACE VEHICLE/ SPACECRAFT

66N32538# ISSUE 18 PAGE 3564 CATEGORY 15 66/04/00 125 PAGES
UNCLASSIFIED DOCUMENT

LUBRICATION EVALUATION
(EVALUATION OF PERFORMANCE OF VARIOUS LUBRICANTS ON BALL BEARINGS
OPERATING IN SIMULATED SPACE ENVIRONMENT)

A/CLAUSS, F. J.

LOCKHEED MISSILES AND SPACE CO., SUNNYVALE, CALIF.

APR. 1966 125 P REFS

/*BALL BEARING/*ENVIRONMENTAL TESTING/*LUBRICANT/*SPACE
ENVIRONMENTAL LUBRICATION/ ADHESION/ BALL/ BEARING/ COLD/ ENVIRONMENT/
FAILURE/ FILM/ FRICTION/ GREASE/ LUBRICATION/ METAL/ OIL/ PERFORMANCE/
SOLID/ SPACE/ TEFLON/ TESTING/ VACUUM/ VISCOSITY/ WELDING

66N26654*# ISSUE 14 PAGE 2606 CATEGORY 15 NASA-CR-75119
E6000-32 NAS8-5251 66/00/00 15 PAGES UNCLASSIFIED DOCUMENT
ENVIRONMENTAL STUDY OF MINIATURE SLIP RINGS QUARTERLY REPORT NO. 11,

1 JAN. - 1 APR. 1966

(SUBLIMATION TECHNIQUE TO LUBRICATE SLIP RINGS IN HIGH VACUUM)

A/KURITZA, O. M.

IIT RESEARCH INST., CHICAGO, ILL. (TECHNOLOGY CENTER.)

AVAIL. NTIS

<1966< 15 P

/*LUBRICATION/*RING/*SUBLIMATION/ ATMOSPHERE/ BRUSH/ GOLD/ HIGH
VACUUM/ ION/ MINIATURE/ NICKEL/ NITROGEN/ NOISE/ OVERLAY/ PUMP/ SLIP/
SPECTROGRAPHY/ TECHNIQUE

66N15241*# ISSUE 6 PAGE 894 CATEGORY 15 NASA-TM-X-54967
65/00/00 15 PAGES UNCLASSIFIED DOCUMENT
IN-FLIGHT TEST TO DETERMINE SPACE ENVIRONMENTAL EFFECTS ON FRICTION,
WEAR, AND LUBRICATION OF MATERIALS

(IN-FLIGHT TEST TO DETERMINE SPACE ENVIRONMENTAL EFFECTS ON
FRICTION, WEAR, AND LUBRICATION OF MATERIALS - PROPOSED MODULAR SYSTEM)

A/DEVINE, E. J.; B/EVANS, H. E.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. GODDARD SPACE FLIGHT
CENTER, GREENBELT, MD. AVAIL. NTIS

<1965< 15 P PRESENTED AT THE AIAA 20 ANN. MEETING AND TECH.
DISPLAY, SAN FRANCISCO, 26-29 JUL. 1965 SUBMITTED FOR PUBLICATION

/*LUBRICATION/*SLIDING FRICTION/*SPACE ENVIRONMENT/*WEAR/ BEARING/
ENVIRONMENT/ FRICTION/ MODULE/ PRESSURE/ ROLLING/ SLIDING/ SPACE/ TEST/
TRANSDUCER/ WELDING

65N22557*# ISSUE 12 PAGE 2028 CATEGORY 15 NASA-CR-62282 K-6055,
PT. II NAS8-11014 65/01/00 50 PAGES UNCLASSIFIED DOCUMENT

FEASIBILITY STUDY OF TECHNIQUES TO PROTECT MECHANISMS OPERATING IN
SPACE FROM MALFUNCTION. PART II- EXPERIMENTAL RESULTS AND
RECOMMENDATIONS FINAL REPORT, 8 JUN. 1963 - 27 JUN. 1964

(VARIOUS TECHNIQUES USED TO PROTECT MECHANISMS OPERATING IN SPACE
FROM MALFUNCTIONS DUE TO FRICTION)

A/JAMISON, W. E.

IIT RESEARCH INST., CHICAGO, ILL. (MECHANICAL ENGINEERING DIV.)
AVAIL. NTIS

JAN. 1965 50 P REFS

/*FRICTION/*SPACE ENVIRONMENTAL LUBRICATION/ ADSORPTION/ CONTACT/
CRYSTAL/ DIFFRACTION/ GAS/ INTERFACE/ LATTICE/ MASS/ MECHANISM/
POTENTIAL/ RESIDUAL/ SPECTROMETER/ TECHNIQUE/ X-RAY

65N22556*# ISSUE 12 PAGE 2028 CATEGORY 15 NASA-CR-62281 K-6055,
PT. I NAS8-11014 64/11/30 88 PAGES UNCLASSIFIED DOCUMENT

FEASIBILITY STUDY OF TECHNIQUES TO PROTECT MECHANISMS OPERATING IN
SPACE FROM MALFUNCTION. PART I- SURVEY AND ANALYSIS FINAL REPORT, 28
JUN. 1963 - 27 JUN. 1964

(SURVEY OF LUBRICATION DEVICES FOR REDUCING FRICTION IN MECHANISMS
OPERATING IN SPACE)

A/JAMISON, W. E.

IIT RESEARCH INST., CHICAGO, ILL. (MECHANICAL ENGINEERING DIV.)
AVAIL. NTIS

30 NOV. 1964 88 P REFS

/*FRICTION REDUCTION/*LUBRICATION SYSTEM/*SPACE ENVIRONMENTAL
LUBRICATION/ ANALYSIS/ ANTIFRICTION/ DESIGN/ ENVIRONMENT/ FRICTION/
LITERATURE/ LUBRICATION/ MECHANISM/ PARAMETER/ REDUCTION/ SPACE/
SURVEY

65N19527# ISSUE 9 PAGE 1476 CATEGORY 15 AEDC-TR-65-35 AD-458414
AF 40/600/-1000 65/03/00 45 PAGES UNCLASSIFIED DOCUMENT

TESTS OF DRY COMPOSITE LUBRICATED BEARINGS FOR USE IN AN AEROSPACE
ENVIRONMENTAL CHAMBER, MAY 1 - JUNE 16, 1964

(DRY COMPOSITE LUBRICATED BEARINGS FOR AEROSPACE ENVIRONMENTAL
CHAMBER)

A/RIDINGS, T. L.

ARO, INC., ARNOLD AIR FORCE STATION, TENN.

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/*BEARING/*ENVIRONMENTAL CHAMBER/*LUBRICATION/*SPACE ENVIRONMENTAL
LUBRICATION/ AEROSPACE/ ALLOY/ BALL/ CHAMBER/ COMPOSITE/ COPPER/ DRY/
ENVIRONMENT/ FACILITY/ GREASE/ LOW/ LOW TEMPERATURE/ PRESSURE/ ROLLER/
SILVER/ SPACE/ TAPER/ TEFLON/ TEST/ TORQUE/ TUNGSTEN/ VAPOR

65N17697# ISSUE 8 PAGE 1284 CATEGORY 15 RR-8-3008 64/02/11
14 PAGES UNCLASSIFIED DOCUMENT

TYPE II FABROID 181 GLASS BACKED SIMULATED OUTER SPACE ENVIRONMENT
QUALIFICATION TEST

(ADAPTABILITY OF FABROID LOW FRICTION, SELF- LUBRICATING BEARING
MATERIAL FOR OUTER SPACE ENVIRONMENT APPLICATION)

A/WAHLBERG, A.

TRANSPORT DYNAMICS, INC., SANTA ANA, CALIF. (RESEARCH AND
DEVELOPMENT DIV.)

11 FEB. 1964 14 P

/*BEARING/*SELF-LUBRICATING MATERIAL/*SPACE ENVIRONMENTAL
LUBRICATION/ ADAPTATION/ APPLICATION/ ENVIRONMENT/ FABROID/ HIGH
VACUUM/ LOAD/ LOW FRICTION/ LUBRICATION/ MATERIAL/ OUTER/ SIMULATION/
SPACE

65N17429# ISSUE 7 PAGE 1086 CATEGORY 15 AEDC-TR-65-1 AD-456990
AF 40/600/-1000 65/02/00 54 PAGES UNCLASSIFIED DOCUMENT

OPERATIONAL EVALUATION OF DRY THIN FILM LUBRICATED BEARINGS AND
GEARS FOR USE IN AEROSPACE ENVIRONMENTAL CHAMBERS, DECEMBER 7, 1963 -
JANUARY 27, 1964

(DRY THIN-FILM LUBRICANTS AND SOFT-METAL LUBRICANTS APPLIED TO
BEARINGS AND GEARS FOR USE UNDER HEAVY LOADS AND SLOW SPEEDS IN SPACE
ENVIRONMENTS)

A/RIDINGS, T. L.

ARO, INC., ARNOLD AIR FORCE STATION, TENN.

ARNOLD ENG. DEVELOP. CENTER, FEB. 1965 54 P REFS

/*BEARING/*GEAR/*LUBRICANT/*SPACE ENVIRONMENTAL LUBRICATION/
ENVIRONMENT/ FILM/ GRAPHITE/ HEAVY/ LOAD/ LOW SPEED/ LUBRICATION/
METAL/ SODIUM/ SOFT/ SPACE/ THIN

65N17276*# ISSUE 7 PAGE 1085 CATEGORY 15 NASA-CR-60783
NAS8-1540 63/00/00 12 PAGES UNCLASSIFIED DOCUMENT

DEVELOPMENT OF SOLID FILM LUBRICANTS FOR USE IN SPACE ENVIRONMENTS
(FRICTION COEFFICIENTS OF INORGANIC SOLID FILM LUBRICANTS FOR USE IN
SPACE ENVIRONMENTS)

A/GADDIS, D.; B/HOPKINS, V.

MIDWEST RESEARCH INST., KANSAS CITY, MO. AVAIL. NTIS

<1963< 12 P PRESENTED AT 1963 USAF AEROSPACE FLUIDS AND
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FILM/ FRICTION/ LAUNCH/ LOAD/ LUBRICANT/ NUCLEUS/ PRESSURE/ RADIATION/
REENTRY/ SOLID/ SPACE/ SPACE FLIGHT/ SPACE VI
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STATE-OF-THE ART APPRAISAL USAF-SWRI AEROSPACE BEARING CONFERENCE,
SAN ANTONIO, TEXAS, MARCH 25-27, 1964
(HIGH TEMPERATURE LUBRICANTS FOR ROLLER BEARINGS USED IN SPACE
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A/WELLONS, F. W.
SKF INDUSTRIES, INC., PHILADELPHIA, PA.
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ROLLER/ SPACE

64N30495# ISSUE 22 CATEGORY 17 FL-210/48 AD-439678 62/12/00 22
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AEROSPACE LUBRICATION RESEARCH SURVEY
(SURVEY OF RESEARCH CONDUCTED ON PROBLEMS OF SPACE ENVIRONMENTAL
LUBRICATION)
A/BROOKS, D. B.
DEFENSE DEPT., WASHINGTON, D.C. (OFFICE OF THE DIRECTOR OF DEFENSE
RESEARCH AND ENGINEERING) AVAIL. NTIS
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LUBRICATION/ OXIDATION/ RESEARCH/ SOLID/ SPACE/ SPACE ENVIRONMENTAL
LUBRICATION/ SURVEY/ VACUUM

64N29508*# - ISSUE 21 CATEGORY 17 NASA-CR-58704 JPL-N2-150168
62/02/16 160 PAGES UNCLASSIFIED DOCUMENT
SURVEY OF FRICTIONAL PROBLEMS IN SPACECRAFT FINAL REPORT
(FRICTIONAL PROBLEMS IN SPACECRAFT MECHANISMS CAUSED BY SPACE
ENVIRONMENTS)
A/IRWIN, A. S.; B/JOHNSON, J. H.
MARLIN-ROCKWELL CORP., JAMESTOWN, N.Y. (RESEARCH AND DEVELOPMENT
LABS.) AVAIL. NTIS
16 FEB. 1964 160 P REFS
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BEARING/ ENVIRONMENT/ GEAR/ LUBRICATION/ MECHANISM/ SEAL/ SPACE/ SPACE
ENVIRONMENTAL LUBRICATION/ SPACECRAFT

64N27310*# ISSUE 19 CATEGORY 17 NASA-CR-58039 NAS8-1540
64/05/22 49 PAGES UNCLASSIFIED DOCUMENT
RESEARCH ON BEARING LUBRICANTS FOR USE IN HIGH VACUUM ANNUAL
SUMMARY REPORT, 23 APR. 1963 - 22 MAY 1964
(SOLID LUBRICANT FOR BEARINGS IN HIGH VACUUM ENVIRONMENT)
A/GADDIS, D. H.; B/HOLM, F. W.; C/HOPKINS, V.; D/HUBBELL, R. D.
MIDWEST RESEARCH INST., KANSAS CITY, MO. AVAIL. NTIS
5 MAY 1964 49 P REFS
/*BEARING/*LUBRICATION/*SOLID LUBRICANT/ BINDER/ COEFFICIENT/
ENVIRONMENT/ FABRICATION/ FILM/ FLUORIDE/ FRICTION/ GOLD/ GRAPHITE/
HIGH VACUUM/ LUBRICANT/ POTASSIUM/ SILICATE/ SODIUM/ SOLID/ SPACE

64N20577*# ISSUE 13 CATEGORY 17 NASA-TM-X-52031 64/00/00 24
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LUBRICANTS AND MECHANICAL COMPONENTS OF LUBRICATION SYSTEMS FOR A
SPACE ENVIRONMENT
(LUBRICANTS AND MECHANICAL COMPONENTS OF LUBRICATION SYSTEM FOR
SPACE ENVIRONMENT)
A/BUCKLEY, D. H.; B/JOHNSON, R. L.
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CENTER, CLEVELAND, OHIO. AVAIL. NTIS
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GEAR/ LUBRICANT/ LUBRICATION/ MECHANICAL/ ROLLING/ SEAL/ SPACE/ SPACE
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64N19952# ISSUE 12 CATEGORY 17 62/02/00 122 PAGES UNCLASSIFIED
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LUBRICATION IN SPACE
(SPACE ENVIRONMENT LUBRICATION)
A/SALMON, W. A.; B/SCAMMELL, H. (SCAMMELL, H. /LITTLE /ARTHUR
D., INC./ ED.)
BAIRD-ATOMIC, INC., CAMBRIDGE, MASS.; LITTLE (ARTHUR D.), INC.,
CAMBRIDGE, MASS.
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/*LUBRICATION/ BEARING/ CHAMBER/ GEAR/ INTERPLANETARY/ PLASMA/
RADIATION/ RING/ SLIP/ SPACE ENVIRONMENTAL LUBRICATION/ SPACE FLIGHT/
SUN/ VACUUM

64N19364* ISSUE 12 CATEGORY 17 64/00/00 6 PAGES UNCLASSIFIED DOCUMENT

ADAPTATION OF A MOS SUB 2 "IN SITU" PROCESS FOR LUBRICATING SPACECRAFT MECHANICAL COMPONENTS

(ADAPTATION OF MOLYBDENUM SULFIDE IN SITU PROCESS FOR LUBRICATING SPACECRAFT MECHANICAL COMPONENTS)

A/VEST, C. E.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. GODDARD SPACE FLIGHT CENTER, GREENBELT, MD.

IN AIAA 5TH ANN. STRUCT. AND MATER. CONF. 1964 P 120-125 REFS /SEE N64-19359 12-01/ AIAA- \$7.50 MEMBERS, \$15.00 NONMEMBERS

/*LUBRICATION/*MOLYBDENUM SULFIDE/*SPACECRAFT COMPONENT/ COMPONENT/ DISULFIDE/ ELECTRODEPOSITION/ ENVIRONMENT/ FILM/ OXIDE/ SPACE/ SPACE ENVIRONMENTAL LUBRICATION/ SPACECRAFT/ SUBSTRATE/ SURFACE

64N17565* ISSUE 9 CATEGORY 17 NASA-RP-146 64/00/00 10 PAGES UNCLASSIFIED DOCUMENT

MECHANISM OF LUBRICATION FOR SOLID CARBON MATERIALS IN VACUUM TO 10 TO MINUS 9 MILLIMETER OF MERCURY

(HIGH VACUUM LUBRICATION OF SOLID CARBON MATERIALS)

A/BUCKLEY, D. H.; B/JOHNSON, R. L.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION. LEWIS RESEARCH CENTER, CLEVELAND, OHIO.

REPR. FROM ASLE TRANS., V. 7, 1964 P 91-100 REFS PRESENTED AT THE LUBRICATION CONF., ROCHESTER, N.Y., 15-17 OCT. 1963

/*CARBON/*HIGH VACUUM/*LUBRICATION/ ALUMINUM/ FRICTION/ MATERIAL/ METAL/ OXIDE/ SOLID/ VACUUM/ WEAR

63N22954# ISSUE 23 CATEGORY 17 AEDC-TDR-63-154 63/10/00 45 PAGES UNCLASSIFIED DOCUMENT

(SPACE ENVIRONMENT LUBRICATION REQUIREMENTS)

A/MC RAE, W. F.; B/PINSON, J. D.

ARNOLD ENGINEERING DEVELOPMENT CENTER, ARNOLD AIR FORCE STATION, TENN.

ARNOLD ENGINEERING DEVELOPMENT CENTER, ARNOLD AIR FORCE STATION, TENN. LUBRICATION REQUIREMENTS FOR SPACE ENVIRONMENTS J. D. PINSON AND W. F. MC RAE OCT. 1963 45P 16 REFS /AEDC-TDR-63-154/

/*LUBRICATION/*SPACE ENVIRONMENT/ ENVIRONMENT/ FACILITY/ SPACE/ SYSTEM

63N19464# ISSUE 19 CATEGORY 17 ASD-TDR-63-474 AF 33/657/-8240
 63/05/00 140 PAGES UNCLASSIFIED DOCUMENT
 (HIGH VACUUM AND HIGH TEMPERATURE ENVIRONMENT MAGNETIC BEARINGS)
 A/EDGAR, R. F.; B/MC HUGH, J. D.; C/MERCHANT, E. W.; D/TONKS, L.
 GENERAL ELECTRIC CO., SCHENECTADY, N.Y. (ADVANCED TECH. LABS.)
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 FRICTIONLESS/ HIGH TEMPERATURE/ HIGH VACUUM/ LUBRICATION/ MAGNETIC/
 MAGNETISM/ MAGNETOHYDRODYNAMICS/ RADIATION/ SEAL/ SUPPORT/ VEHICLE

63N17683# ISSUE 16 CATEGORY 17 MTI-62TR34 AF 33/657/-8666
 62/11/06 47 PAGES UNCLASSIFIED DOCUMENT
 (COMPLEX BEARING AND/OR LUBRICATION SYSTEMS)
 A/LEWIS, P.; B/MURRAY, S. F.; C/PETERSON, M. B.
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 MECHANICAL TECHNOLOGY INC., LATHAM, N.Y. INVESTIGATION OF COMPLEX
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 CERAMICS/ CIRCULATION/ EQUIPMENT/ FILM/ FLIGHT/ FLUID/ GAS/ HIGH
 TEMPERATURE/ HIGH VACUUM/ LUBRICATION/ MAGNETIC/ MAGNETISM/ METAL/
 MOLYBDENUM/ NUCLEAR/ OXIDE/ RADIATION/ ROLLING/ ROTATION/ SILICATE/
 SOLID/ STELLITE/ SUSPENSION/ SYSTEM/ TITANIUM/ TUNGSTATE

63N17014# ISSUE 15 CATEGORY 33 RM-3164-PR AD-276365 AF
 49/638/-700 62/05/00 11 PAGES UNCLASSIFIED DOCUMENT
 (ORIGIN OF FATIGUE CRACK AS SURFACE PHENOMENA)
 A/SHANLEY, F. R.
 RAND CORP., SANTA MONICA, CALIF.
 RAND CORP., SANTA MONICA, CALIF. NEW EVIDENCE CONCERNING THE
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 DUCTILITY/ EFFECT/ GROWTH/ HIGH VACUUM/ LUBRICATION/ RESISTANCE/ SLIP/
 TENSION

63N15479 ISSUE 12 CATEGORY 18 SB-63-5 63/03/00 UNCLASSIFIED
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A/BUGINAS, S. J.

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MISSILE/ OXIDATION/ REDUCTION/ SEIZURE/ SPACE/ SURFACE/ TEMPERATURE/
VACUUM/ WEAR

63N10929 ISSUE 2 CATEGORY 17 62/00/00 12 PAGES UNCLASSIFIED
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(LUBRICATION IN SPACE ENVIRONMENTS)

A/ADAMCZAK, R. L.; B/BENZING, R. J.; C/SCHWENKER, H. S.

DIRECTORATE OF MATERIALS AND PROCESSES, AERONAUTICAL SYSTEMS DIV.,
WRIGHT-PATTERSON AFB, OHIO.

DIRECTORATE OF MATERIALS AND PROCESSES, AERONAUTICAL SYSTEMS
DIV., WRIGHT-PATTERSON AFB, OHIO LUBRICATION IN SPACE ENVIRONMENTS R.
L. ADAMCZAK, R. J. BENZING, AND H. SCHWENKER IN SOC. OF AEROSPACE
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HYDROCARBON/ LUBRICANT/ MINERAL/ OIL/ POWDER/ PRESSURE/ SILICON/ SPACE/
TRANSFER

62N12809*# ISSUE 8 CATEGORY 18 JPL-TR-32-150
NASW-6 61/11/01 124 PAGES UNCLASSIFIED DOCUMENT

(MATERIALS BEHAVIOR IN SPACE ENVIRONMENT)

A/JAFFE, L. D.; B/RITTENHOUSE, J. B.

JET PROPULSION LAB., CALIF. INST. OF TECH., PASADENA.

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CONFIGURATION/ DEBRIS/ EFFECT/ ENGINE/ ENGINEERING/ ENVIRONMENT/ EPOXY/
EROSION/ ESTER/ LUBRICANT/ LUBRICATION/ MATERIAL/ METEOROID/ NUT/
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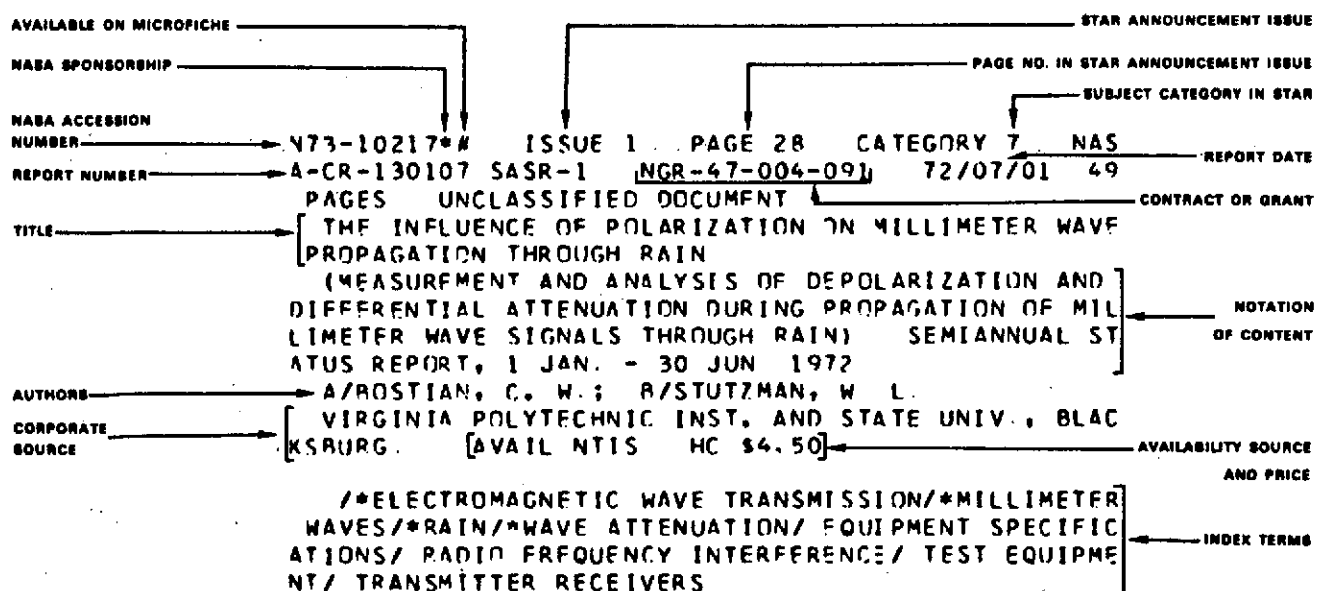


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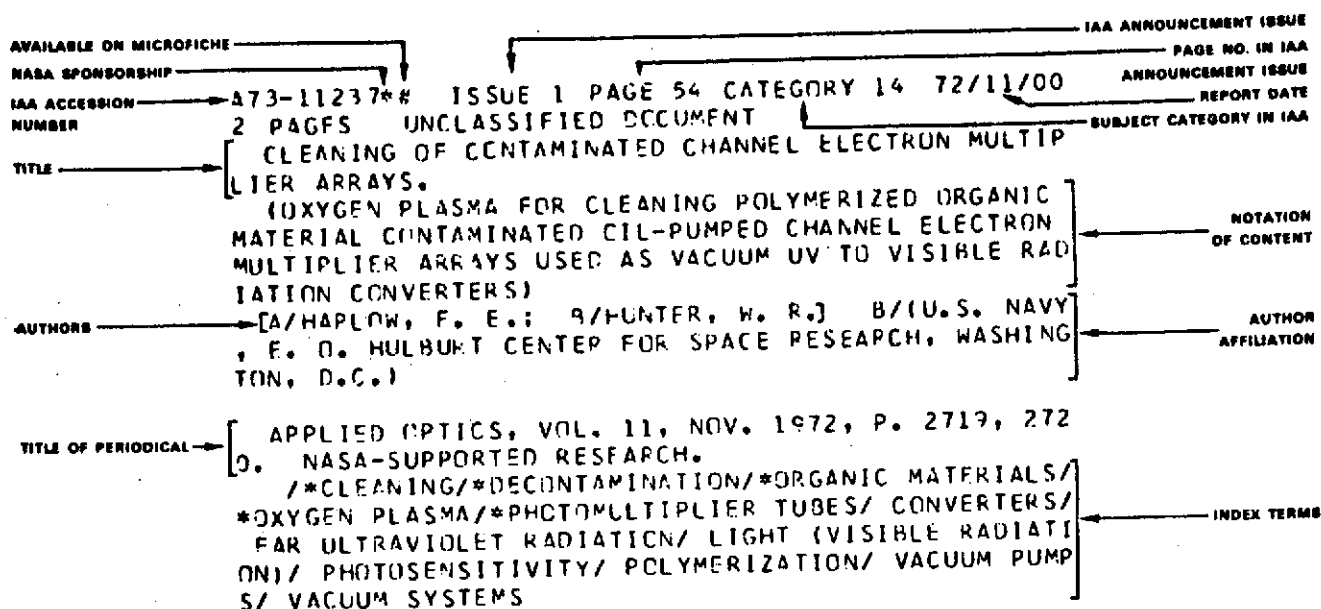


FIGURE 2. PUBLISHED LITERATURE CITATION

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U.S. Geological Survey
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Bldg. 25, Denver Federal Center
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Zentralstelle für Luftfahrt-doku-
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8 München 86
Postfach 880
Federal Republic of Germany

TO: Engineering File MT 15,774
From: C. Gurrisi

Issue: Original
Date: October 25, 1974

ATM CMG LUBRICATION SYSTEM
FLOW RATE TEST

Prepared by: C. Gurrisi
C. Gurrisi

Approved by: R. Abramowitz
R. Abramowitz

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY

Issue: Original
Date: October 25, 1974

MT 15,774
Page 1

PURPOSE

To determine if the flow rate of the ATM CMG Dynamic Lubrication System is sufficient in the temperature range of 50°F to 80°F.

DISCUSSION

As part of the failure investigation for ATM CMG S/N P-5 which failed onboard Skylab, a test program was initiated at Bendix Guidance Systems Division to measure the lubricant flow rate from the control moment gyros inner gimbal and rotor assembly (IGRA) dynamic lubrication system in the temperature range of 50°F to 90°F. Lubrication system flow rate at 90°F was specified to be between .05 and .120 mg/hr with a variance of .030 mg/hr at 90°F, from one test run to the next run. It should be noted that all testing was performed at 8000 rpm, however, the system operates in the IGRA at 9100 rpm. Therefore all measured flow rates will have to be ratioed up by multiplying the measured flow rate at 8000 rpm by $\left(\frac{9100}{8000}\right)^2$ to yield the flow rate at 9100 rpm.

Prior to any testing at low temperatures, three room temperature flow rates were taken on lubrication system S/N 15 to establish a firm baseline set of data. These runs resulted in flow rates of .048, .056 and .051 mg/hr at 8000 rpm and at temperatures of 79.5°F, 82.4°F, and 83°F, respectively. When normalized to 90°F, these flow rates become .075, .077 and .069 mg/hr, respectively, which are well within the specification of .05 to .120 mg/hr and also satisfy the .030 mg/hr allowed variation. When upgraded to 9100 rpm operation, the flow rates are .0975,

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Page 2

.10 and .0896 mg/hr.

The flow rate test fixture with lubrication system S/N 15 was then set up in a standard cabinet cold chamber for testing in the 50°F to 60°F temperature range. The chamber was stabilized at a temperature of 60°F and then a 114 hour flow rate test was performed. At the completion temperature was 67°F. The chamber was then stabilized at 50°F and a 105 hour flow rate test was run, resulting in a lubrication system temperature of 56°F. See Table 1 for the results of these tests.

Once these two temperature exposures were completed, a flow rate test was performed at room temperature to verify the integrity of the lube system. This flow rate test required in a normalized flow rate of .072 mg/hr at 8000 rpm. Since this flow rate agreed perfectly with the room temperature baseline tests, the system was not altered by the cold temperature exposures.

Flow rate tests were then repeated at chamber temperatures of 52°F and 45°F with resulting lube system temperatures of 58°F and 52.5°F, respectively, to verify the results of the first two flow rates at the colder temperatures. Correlation of the first set of cold temperature data to the second set of cold temperature exposures was excellent. At the completion of this second set of cold temperature exposures another room temperature flow rate was performed to again verify the integrity of the lubrication system. This verification flow rate also agreed well with the initial baseline tests.

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To verify the normalization factors to obtain the flow rates at 90°F, a flow rate measurement was performed with a chamber temperature of 85°F and a lubrication system temperature of 88°F for 98.8 hours.

Table 1 presents all the collected data on the previous mentioned tests. This table presents the flow rates at each temperature, the normalized flow rate for each run to 90°F, and the flow rate for each test normalized to 90°F and 9100 rpm.

CONCLUSIONS

The test data compiled during this test program confirms that the lubricant (KG-80 oil) flow rate from the ATM CMG IGRA dynamic lubrication system does decrease with operation at lower operating temperatures. These tests show an ≈18.5% decrease in flow rate between 88°F and 52.5°F.

The compiled data for flow rate test runs between 50°F and 90°F at 8000 rpm have been plotted in Figure 1. Figure 1 also presents the theoretical curves for flow rates of .05 and .120 mg/hr at 90°F from 90°F to 50°F at 8000 rpm. Since these three plots present parallel straight lines the theoretical flow rate temperature compensations on all previous lube nut testing is valid.

Figure #2 is a plot of the test data for lube system S/N 15 plus the theoretical curves of temp vs flow rate for flow rates of .05 and .120 mg/hr at 90°F, adjusted to 9100 rpm. Again the three curves are parallel straight

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Date: October 25, 1974

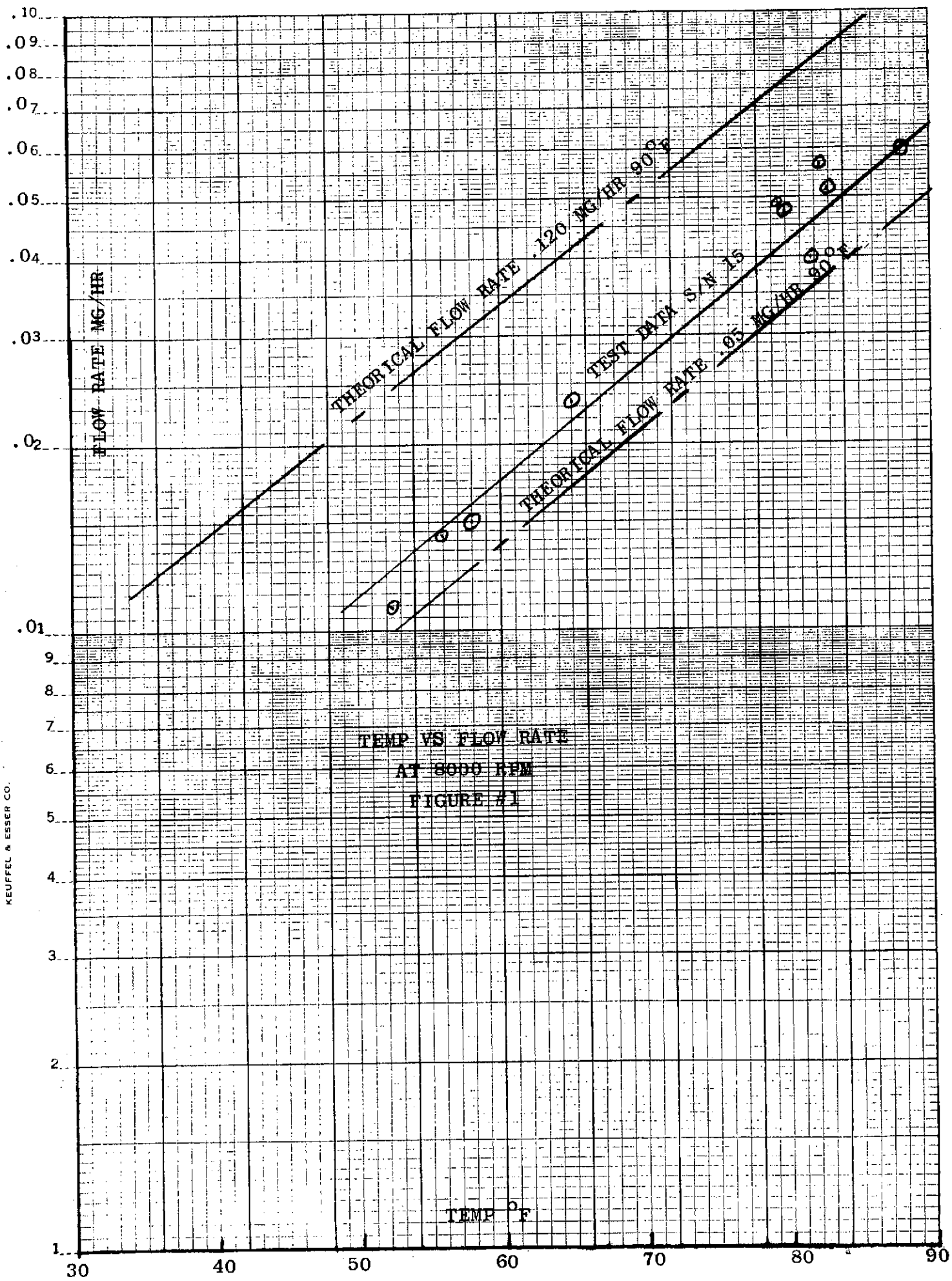
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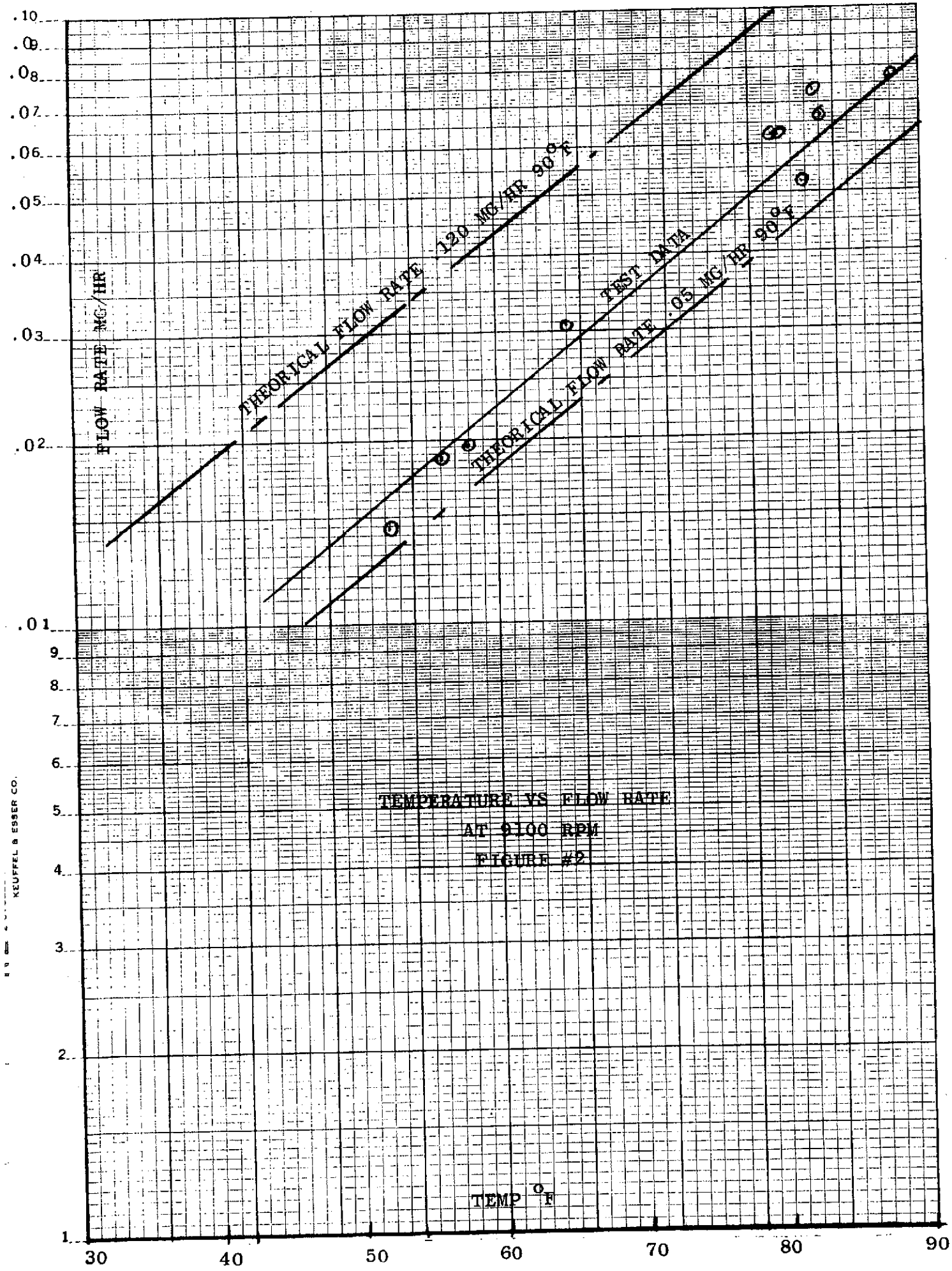
lines which further validates the theoretical flow rate
temperature compensation factors.

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C

LUBRICATION SYSTEM S/N 15 FLOW RATES

CHAMBER TEMP °F	LUBE SYS. TEMP °F	COLLECTOR DRY WT GRMS	COLLECTOR WET WT GRMS	QTY OF OIL FLOW MG	RUN TIME HRS	FLOW RATE AT NUT TEMP °F 8000 RPM MG/HR	FLOW RATE NORMALIZED TO 90°F 8000 RPM MG/HR	FLOW RATE ADJ TO 9100 RPM AT NUT TEMP, MG/HR	FLOW RATE NORMALIZED TO 90°F 9100 RPM MG/HR
N/A	79.5	31.1065	31.1145	8.0	165.3	.048	.075	.0625	.0975
N/A	82.4	39.0645	39.0736	9.1	163	.056	.077	.073	.100
72	83	39.0646	39.0705	5.9	114	.0516	.0695	.067	.0903
60	65	39.0646	39.0673	2.7	114	.0236	.0685	.0307	.089
50	56	39.0644	39.0659	1.5	105	.0143	.0586	.0186	.0761
80	80	39.0644	39.0698	5.4	112.9	.048	.072	.0625	.0935
52	58	39.0645	39.0660	1.5	100.1	.015	.0592	.0195	.0768
45	52.5	39.0644	39.0660	1.6	145.2	.011	.0545	.0143	.071
72	81.8	39.0645	39.0684	3.9	97.5	.040	.0569	.052	.074
85	88	39.0638	39.0697	5.9	98.8	.0597	.060	.0776	.078





TEMPERATURE VS FLOW RATE
AT 9100 RPM
FIGURE #2

KEUFFEL & ESSER CO.

To: Engineering File MT-15,783
From: J. Pona

Issue: Original
Date: 17 December 1974

ATM CMG THERMAL VACUUM TEST

Prepared by:

Joe Pona
J. Pona

Approved by:

R. Abramowitz
R. Abramowitz

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608

Issue: Original

MT-15,783

Date: 17 December 1974

Page 1

ATM CMG THERMAL VACUUM TEST

PURPOSE

To determine the effects of temperature on the drag torque of the CMG spin bearing.

BACKGROUND

The failure of CMG No. 5 (Skylab CMG 1) and the anomalies which occurred with CMG No. 6 (Skylab CMG 2) resulted in the investigation of the effects of temperature on the bearing torque of the CMG spin axis bearings. A life test fixture equipped with two normally lubricated bearings was placed in a thermal vacuum chamber and subjected to temperature changes from 10°F to 155°F. Current, speed, temperature and torque were monitored throughout the test.

DISCUSSION

Life Test Fixture (BLTF) S/N #2 was equipped with two normally lubricated spin bearings. In order to maintain a fixed quantity of oil (KG-80) in the ball bearing, thus eliminating torque variations as a function of lube quantity, the active lubrication systems were not utilized during this test.

All testing was performed in a thermal vacuum chamber (NRC model 2004), see attached ETS 10152, with the unit securely mounted on a thermal platen. A thin layer of

Issue: Original

MT-15,783

Date: 17 December 1974

Page 2

thermal silicon grease was applied to the mating faces of the BLTF and the thermal platen in order to insure proper thermal conductivity. The unit was accelerated to operating speed (9100 RPM) at room temperature ($\approx 70^{\circ}\text{F}$) and hard vacuum ($\approx 10^{-7}$ torr).

A baseline set of data consisting of time, speed, temperature, vibration and bearing torque was taken. This resulted in an initial bearing torque of 1.62 oz-in. The chamber temperature was decreased to 0°F and the unit was allowed to stabilize overnight. After stabilization, the bearing temperatures were 10°F and 12°F on sides 1 and 2 respectively; however, the unit was no longer operating (was at zero speed). Without removing power from the BLTF the chamber was gradually increased in temperature until wheel rotation was detected. Once rotating, acceleration to 9100 RPM occurred in the normal interval of time and a torque test indicated the torque was slightly lower than the baseline torque (1.47 oz-in as opposed to 1.62). The unit temperature was then decreased slowly from 65°F in approximately 10°F increments. Torque tests were taken at each plateau. At 24°F the bearing drag torque exceeded the motor torque and the unit stalled. The temperature was increased in 10°F steps to 155°F with torque tests taken after each stabilization. The final room temperature torque test resulted in a bearing torque of 1.3 oz-in.

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Date: 17 December 1974

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Holding the unit at 10^{-7} torr, two 24 hour temperature cycles were performed. The cycles started at 72°F, was increased to 90°F, decreased to 50°F, and returned to 72°F. Torque tests were taken at the start and finish of both cycles showing ≈ 1.30 oz-in of torque on all occasions.

These temperature cycles were performed in an attempt to produce any of the anomalies noted in CMG #2 on-board Skylab.

CONCLUSIONS

Examining the data and the enclosed graph, we see that the viscous drag increases with decreasing temperature caused by the change in viscosity of the lubricant. The exponential increase in torque below 30°F is due, not only to viscosity change but primarily to the preload change in the BLTF. This change in preload is a result of the fact that the life test fixture configuration does not allow the thru strut to control the bearing preload spring clearance as it does in an actual unit assembly. In the BLTF the aluminum housing with its greater coefficient of expansion and its inherently stiff cylindrical configuration closes down the preload spring clearance at colder temperatures.

From the attached graph we see that the torque decreased consistently from 2.63 oz-in at 30°F to .84 oz-in at 155°F. The variations in torques above 30°F are caused

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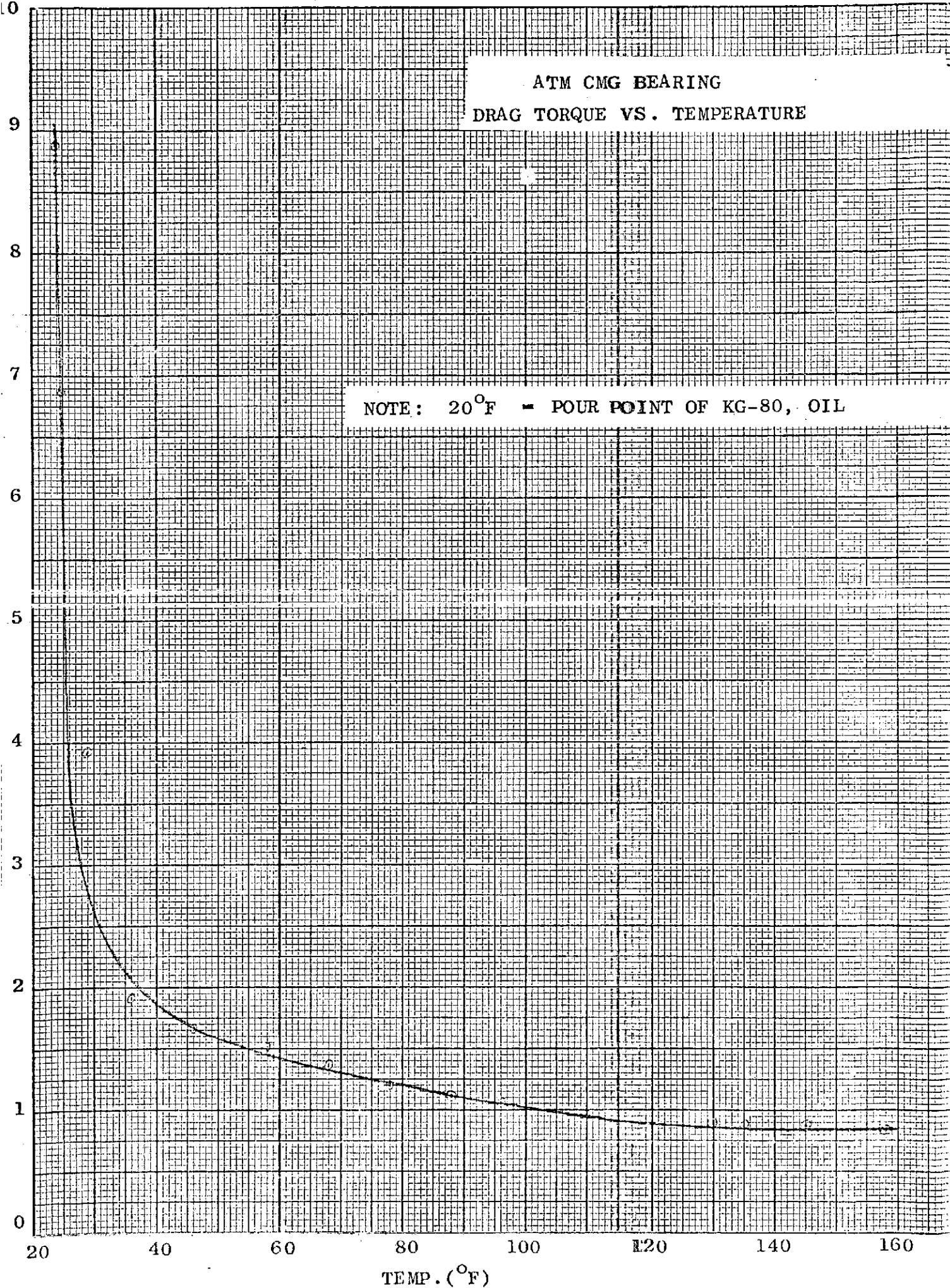
solely by the viscosity change in the oil. The two temperature cycles between 90°F and 50°F showed no variations of current, torque or speed.

Date: December 17, 1974

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ATM CMG BEARING
DRAG TORQUE VS. TEMPERATURE

NOTE: 20°F - POUR POINT OF KG-80, OIL




Issue: Original

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Date: 17 December 1974

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HOURS	1/2 SPEED RPM	TORQUE OZ-IN	BEARING TEMP.		VIBRATION	
			T ₁ °F	T ₂ °F	SIDE 1	SIDE 2
0:00	4525	1.62	70	70	10	7
1:00	4525	2.03	54	56	22	20
18:00	0	-	10	12	-	-
20:35	4528	1.47	64	66	10	9
22:20	4527	1.51	42	44	11	8
23:50	4514	1.92	32	34	12	10
41:30	4492	2.66	30	31	9	8
41:35	4450	3.93	29	30	11	8
41:45	4400	5.07	26	27	10	8
41:50	4350	6.87	25	26	10	10
42:00	3972	8.87	24	25	10	10
43:10	4515	1.91	36	35	15	12
44:50	4521	1.66	46	47	8	8
46:00	4527	1.50	58	58	9	7
48:15	4531	1.37	68	68	8	8
67:45	4533	1.20	78	77	8	8
69:00	4536	1.12	88	85	8	8
137:00	4536	1.06	96	92	16	8
139:30	4536	.95	110	105	15	8
163:30	4537	.90	130	125	19	8
164:30	4537	.89	135	130	19	7
165:30	4539	.87	145	140	15	8
168:00	4540	.84	158	150	11	8
185:00	4530	1.27	70	72	8	8
237:00	4528	1.31	70	73	8	9

TO: C. Gurrisi	 ENGINEERING TEST SUMMARY		ETS NO. 10152	
			PROGRESS FINAL <input checked="" type="checkbox"/>	
			DATE 8/27/74	
COPIES TO: P. Imbeninato T. Sheppard J. Pona	TEST SPECIMEN DESCRIPTION			
	CMG SPIN BEARINGS			
	MFR. OF SPECIMEN BENDIX	MODEL NO. 2123570-1	SERIAL NO.(S) NONE	
	ASSOCIATED PROGRAM OR SYSTEM DESCRIPTION			
	ATM-CMG			
TEST ENGINEER R. Sabato	SUPERVISOR	TYPE OF TEST THERMAL VACUUM	START TEST (DATE) 8/13/74	END TEST (DATE) 8/26/74
		COMPREHENSIVE REPORT FOLLOWS	YES NO	<input checked="" type="checkbox"/>

OBJECT OF TEST:

To evaluate the effect of hard vacuum and temperature on the CMG Spin Bearings - Drag Torque Test.

TEST EQUIPMENT AND IDENTIFICATION	Calib. Due Date	Freq. Calib.
Thermal Vacuum Chamber (NRC Model 2004)	8/31/74	1 yr.
Temperature Indicator (Controller) ALNOR	8/31/74	1 yr.

TEST SET-UP(S):

The specimen was positioned on the thermal platen inside the chamber utilizing an application of thermal silicon grease. The specimen was interfaced to its test stand for functional data generated during testing. A thermocouple junction was located on the platen and was used to control the temperature of the platen. Additional thermocouples were attached to the platen in such a manner as to monitor temperature distribution along the platen surface (RE: Fig. 1).

DISPOSITION OF SPECIMEN(S):

Returned to the Engineering Project Group.



ENGINEERING TEST SUMMARY (CONTINUATION SHEET)

TESTING PERFORMED AND RESULTS:

The specimen was subjected to reduced pressures in the 5×10^{-7} torr range. During this condition, various temperature test points were accomplished between 0°F and 160°F . During these parameters, the specimen was energized and functional data generated is in the possession of the Engineering Project Group.

At the completion of this portion of testing, the specimen was subjected to two 24 hour pressure/temperature cycles as illustrated in Table I. Data generated during these tests was retained by the Engineering Project Group.

At the completion of all testing, the specimen was visually examined and no deterioration was observed. No malfunctions were reported by the Project Group.

Thermal Vacuum Log Sheets are on file in the Thermal Vacuum Laboratory.

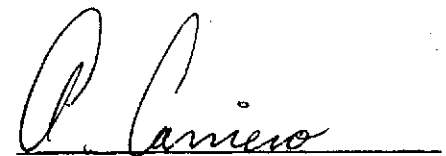
To: Engineering File MT-15,786
From: A. Carriero

Issue: Original
Date: 6 January 1975

SIMULATION OF SKYLAB CMG 2 DATA

MINIMAL BEARING LUBRICATION

Prepared by:



A. Carriero

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY

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Date: 6 January 1975

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1.0 PURPOSE

The purpose of this report was to document the results of tests using minimally lubricated bearings in an attempt to simulate current and speed anomalies similar to those that occurred to the Skylab CMG #1 (S/N 5) and CMG #2 (S/N 6) during the Skylab mission.

2.0 BACKGROUND

The primary mechanisms used to orient and maneuver the Skylab space laboratory were three Control Moment Gyros manufactured at the Bendix Guidance Systems Division. These CMG's used two 107 H size angular contact ball bearings, each had a wheel weight of 150 pounds, ran at a speed of 9100 RPM and developed an angular momentum of 2300 ft-lb-sec. The CMG's were tested both at Bendix and at NASA Huntsville for approximately 1500 hours prior to their launch in the Skylab space laboratory on May 14, 1973.

On November 23, 1973, after 195 days of continuous operation in space, ATM CMG S/N 5 (Skylab #1) experience what appeared to be a bearing failure and ceased to function. During the remaining 75 days of the mission, close monitoring of the speed, current and temperature data from the remaining two functioning ATM-CMG's revealed periods of apparent unstable behavior in ATM-CMG S/N 6 (Skylab #2). These anomalies were in the form of slight speed, current, and bearing temperature changes from normal,

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but did not affect the functioning of the gyro. However, the trend of the data indicated a possible deterioration in the performance of one of the two bearings paired in this gyro. None of these anomalies were noted in the data obtained from ATM CMG S/N 7, (Skylab #3). Both of these remaining CMG's completed the mission of approximately 6500 hours in space and were run down after the egress of the Skylab III astronauts on February 8, 1974.

The speed, current and temperature changes noted were known as "glices" and lasted anywhere from several minutes to several hours. "Glices" which remained constant in both current and speed for most of its duration were referred to as "plateaus". Some plateaus were of several hours duration. At no period, however, did the wheel speed decrease more than 120 RPM below the nominal speed (not necessarily 9100 RPM). This was possibly due to telemetering and instrumentation variations as well as data collection location differences. There were also other periods of "un-glices" and "un-plateaus" where the speed appeared slightly higher than nominal and the current was correspondingly lower.

Other factors such as temperature cycling, gravity gradient desaturations, and maneuvers which may have directly or indirectly affected the initiation or termination of an anomaly period. A change in the bearing cartridge heater temperature cycle during the last seven weeks of the mission led to a prolonged

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period, (four and one-half days) during which no anomalies were observed. During the first 32 weeks of the mission the gyro temperature of CMG S/N 6 (Skylab #2) cycled between $+60^{\circ}$ and $+80^{\circ}\text{F}$. During the last seven weeks the cycle was $+72^{\circ}$ to $+78^{\circ}\text{F}$, thereby increasing the lube nut (oil replacement system) flow rate, which increased the quantity of oil to each bearing and lowered the viscosity of the oil. The effect of this change was to decrease the frequency of occurrence of the "glices" at least temporarily. It was during this later period of increased average temperature that several periods of "unglices" occurred.

A review and investigation of this data by a committee of personnel from various NASA centers, Bendix engineering, as well as independent consultants led to the conclusion that primary factor for the occurrence of these anomalies was most likely lack of adequate lubrication in the #2 bearing on CMG #6. (Ref. 40M23157 NASA-MSFC). Therefore the tests in this study were conducted utilizing bearings with known degrees of marginal lubrication in an attempt to simulate the temperature, speed and current anomalies experienced by CMG S/N 6 (Skylab #2) and CMG S/N 5 (Skylab #1).

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3.0 DISCUSSION

Three tests were conducted under the following conditions of bearing marginal lubrication.

a) Test # 1

Dry retainer
Dry bearing metal parts

b) Test #2

Dry retainer
500-1 solution of freon to KG-80 oil on the bearing metal parts

c) Test # 3

Dry retainer
50-1 solution of freon to KG-80 oil on the bearing metal parts.

All of these tests were performed in the spin axis vertical position with the test bearing at the bottom. The tests were conducted in this attitude to simulate a worst case condition where the oil from the lube nut must migrate or be pumped against 1 "g" to relubricate the minimally lubricated test bearings. It is assumed that in a zero "g" space environment, the dispersion and migration of the oil will be more random than in a 1 "g" field. Assuming similar temperature gradients,

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the space environment would allow quantities in excess of these tests to enter the bearing race contact area. Therefore, if sufficient oil migrates against 1 "g" to replenish the minimally lubricated test bearings in these tests, then there is greater probability that the Skylab bearings in question were adequately relubricated. If the test bearings do not recover sufficient oil, there is more reason to doubt the sufficiency of the rate of oil migration into the bearing race of the Skylab bearings.

The bearing used was Barden S/N BB18, a bearing that had been previously run on life tests for approximately 1000 hours, and which exhibited the normal metal tracking associated with this amount of run-in time.

In each test the bearing metal parts were cleaned per MT-13,923 after removal of whatever residues and debris were left from previous tests.

The bearing retainers used in each test were completely devoid of oil to prevent supplementary lubrication of the bearings. This was accomplished by extraction in freon for approximately 16 hours and vacuum baked for twelve hours prior to assembly into the test bearing. The test bearing was then assembled along with a normally lubricated bearing into a life test fixture and preloaded to the normal 40# thrust load as in the Skylab ATM-CMG's. The life test fixture applies

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no radial load other than the weight of the shaft and so simulates to a certain extent the weightlessness of the ATM-CMG wheel in space.

The life test fixture is driven by one standard ATM-CMG motor developing approximately nine oz-in. of torque at 455 Hz and 130 volts. Synchronous speed is approximately 9100 RPM. The normal torque of bearing paired in a life test fixture is 1.5-2.0 oz-in. Since there is very little inertia for the motor to contend with in the life test fixture, small torque changes will be reflected in pronounced speed and current changes. The inertia of the life test fixture shaft is .045 in. lb. sec² compared with 29.0 in. lb. sec² for an ATM-CMG.

Various test equipment was set up to monitor these changes. (See Photo 1). Visual readout of speed, motor current, vibration levels of both bearings and the temperatures of the test bearing was provided. In addition, a six channel recorder was continuously monitoring A ϕ motor current, wheel speed (actually 1/2 speed), test bearing temperature, and the vibration levels of both bearings. However no meaningful data was recorded during test number 1 since the test bearing "failed" within three minutes. The speed traces recorded during tests 2 and 3 utilized an automatic reset device. During test number 2 the pen reset every ten counts, or 20 RPM, which made for very accurate recording of small speed changes. However, larger speed changes were very

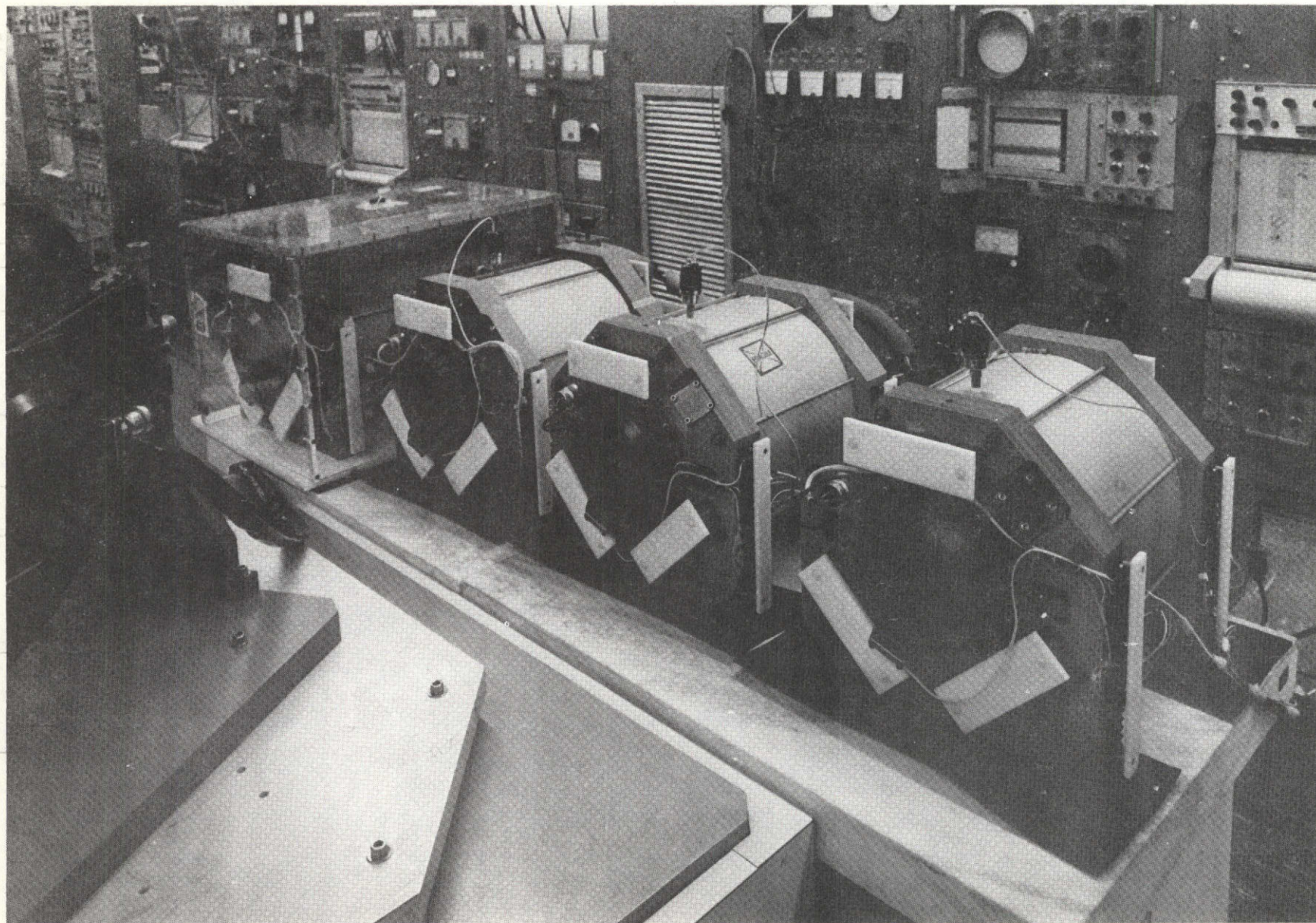


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difficult to interpret without visually reading the counter. During test number 3 the setting was changed to reset at 100 counts or 200 RPM intervals. Sudden speed changes in excess of 200 counts were difficult to interpret on the tape and many speed readings taken during "failure" and "recovery" periods were visual readings from the speed counter.

4.0 RESULTS

Test #1 - The completely dry bearing of Test #1 ran for approximately three minutes, including a run up time of approximately 45 seconds. The bearing torque then exceeded the motor torque and the speed dropped to zero. The period of run was too short to record any meaningful data.

Visual examination of the bearing metal parts and retainer revealed that the increase in torque was caused by the deposition of residues and debris from the bearing retainer pockets onto the bearing races and balls. This debris and resinous residues deposited on the races and balls was generated by the friction and localized overheating caused by the rub of the balls in the lead and lag segments of the unlubricated ball pocket surfaces.

The bearing metal parts showed no wear and no evidence of spalling or oxidation after removal of the debris and residues. The retainer was ultrasonically recleaned in

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freon and except for the burnishing in the ball pockets which occurred during the test, there was no other physical damage. The retainer was then vacuum baked for 16 hours at 180°F.

Test #2 was conducted using the same bearing and retainer as in Test #1. In this test the metal parts were dipped into a 500-1 mixture of freon to KG-80 oil, leaving a very thin film of oil on the metal parts after evaporation of the freon. The bearing was then assembled with the cleaned and dried retainer and replaced into the life test fixture.

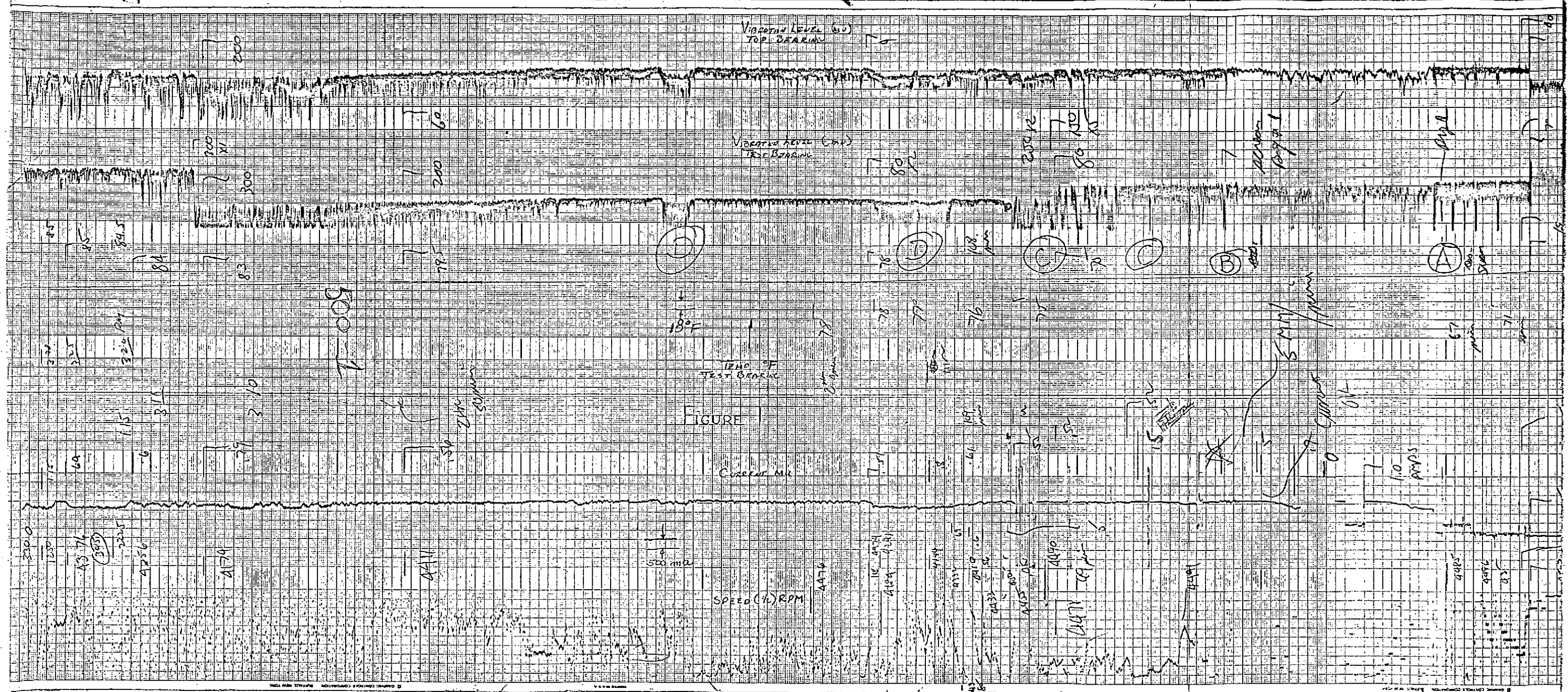
Test #2 ran for approximately eleven minutes before anomalies in the traces were noted. At this point in the test, calibration of the test equipment was still in progress, and direct correlations could not be made accurately. However, for the next several minutes there were periods of audible retainer squeal, accompanied by speed and current changes.

The remainder of the 3-1/2 hour run illustrated a variety of anomalies (see Figure 1). There were cases of sharp changes in vibration level both with and without changes in speed or current. There were periods of change in speed and current both with and without changes in vibration level. The period of audible bearing, or retainer noise, always corresponded to changes in vibration level but not always to speed and current changes.

ORIGINAL
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FOLDOUT FRAME

FOLDOUT FRAME

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The point marked A on the trace (Figure 1) was an example of audible retainer noise with a corresponding significant current increase and speed decrease. Point B was another instance of audible retainer noise but without a corresponding current increase or speed decrease. The area of the trace between the letters C contained points where the vibration levels increased and decreased sharply accompanied by drastic speed and current changes. Speed changes up to 1000 RPM were noted and current changes of over 600 ma were recorded. Area D indicated a more moderate increase in vibration level with corresponding 200-300 ma. increases in current and 150-700 RPM speed decreases.

After the second area D there was a general recovery of the bearing after which there was a gradual deterioration indicated by a steady increase in overall vibration level, motor current, and a decrease in speed. There was also a gradual, then accelerated increase in bearing temperature to +84°F from +73°F. During this one hour segment there were periods of deterioration and recovery involving changes of 100-300 ma. and several hundred RPM.

At approximately 3:20 PM the bearing stopped momentarily then ran at approximately 4450 RPM (2225 counts x 2) for several minutes. The speed then increased to approximately 9050 RPM for a minute, dropped to 2500 RPM, recovered, then stopped and the test was terminated.

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The first sustained "plateau" occurs at the first "D" area. The current rose from 510 ma to 610 ma and then to 800 ma. The current then dropped to about 650 ma occurring with a corresponding decrease in the vibration level. The plateau then remained at approximately 700 ma for the balance of this period. No audible retainer noise was noted during this period. The probable cause for the vibration level and torque increase was deposition of resinous residues from the retainer ball pockets into the ball path. This plateau is several times the magnitudes noted with Skylab CMG #2 (S/N 6). This plateau may be compared with the plateau of DOY 323:16 - 323:10, Figure 2. In both cases there is an increase in current and a corresponding increase in temperature in both the test bearing and bearing #2 in the CMG. However, the decrease in speed of 660 RPM during this test period represents a torque increase of 7-8 oz-in., while the apparent change in the speed of ATM CMG #2 was in the neighborhood of 30-40 RPM or about 1 oz-in. It may also be noted that the temperature modes are dissimilar. The test bearing was approaching "failure" as the temperature was still ramping higher after the end of the "plateau". The temperature of CMG S/N 2, bearing #2 returned gradually to its former level (measured as a Δ from bearing #1).

After termination of test #2 the test bearing was removed from the life test fixture. The bearing was disassembled and the bearing metal parts and retainer

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visually examined. Examination indicated that the primary cause for the torque increase was the build-up of debris and residues on the bearing balls and races. There was more defined wear in the bearing retainer ball pockets which also bore evidence of retainer instability.

The retainer flange onto which oil is centrifuged from the lube nut contained no visible evidence of oil. Whatever oil had been centrifuged onto the flange had been in turn centrifuged thru the feed holes and onto the bearing outer race land.

There was also evidence of wear on the retainer scallop pads on the pressure side of the inner race. There was evidence of burnishing on the pressure side of the inner race land (see Figure 3) indicating that the 500-1 solution (freon to oil) is insufficient to adequately lubricate the retainer pads and the area where the retainer pads contact the inner race land.

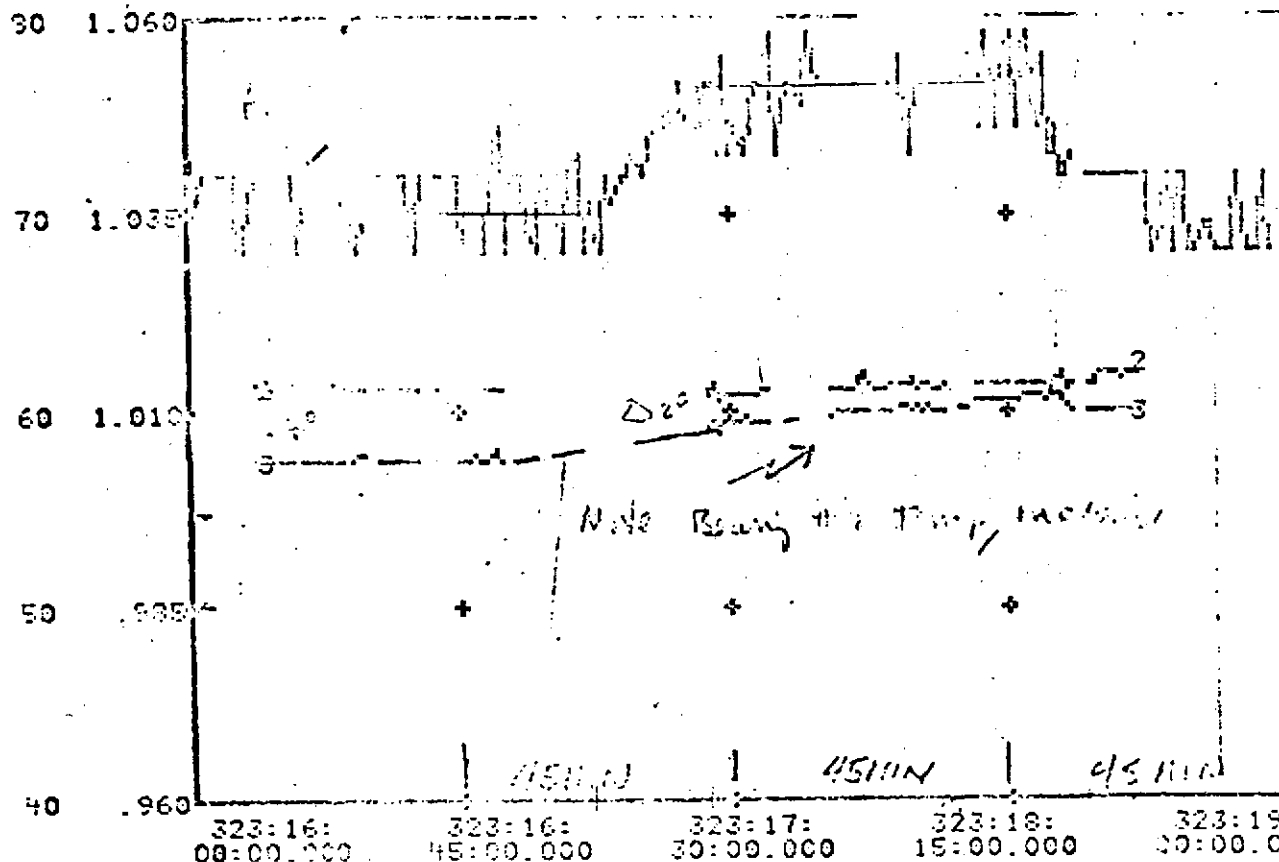
The ball pockets had considerable wear in both the lead and lag portions, and there was evidence of burnishing on the sides of the pockets, perpendicular to the direction of motion, indicating periods of retainer instability.

The bearing races contained loose retainer debris, ground in resinous residue, and minute amounts of metal wear products.

REP-0540. 31-0113 VIEW10-HV1 0002 TO 0002 OF 0003
SWS TRANSFER BUS CURRENTS

XS: GMT
EV: EVT0100L

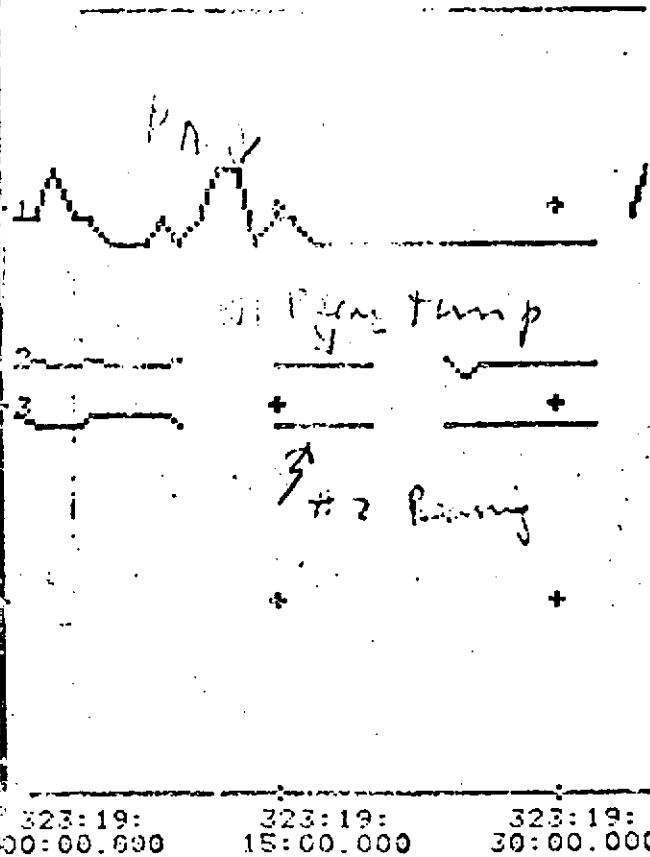
NHL A AMP BSG 1 T F BSG 2 T F
00007-702 =1(R) 00008-702 =2(L) 00009-702 =3(L)



31-0113 VIEW10-HV1 0003-0002 TO 0003-0002 OF 0003
SWS TRANSFER BUS CURRENTS

XS:
EV:

BSG 1 T F BSG 2 T F
(R) 00008-702 =2(L) 00009-702 =3(L)



DAMP
FIGURE 2

LS1E2W03

L15

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The bearing race contact areas when cleaned showed no evidence of metal distress; i.e., spalling, oxidation. The contact wear areas appeared slightly wider than prior to the test run. The balls exhibited an additional set of fairly wide wear bands caused by running in a preferential orientation for a period of time during the test. However there was no apparent metal deterioration.

The area of the outer race which was covered by fine loose debris from the retainer pockets extends from a line on the outer race land just inboard from the edge of the retainer, upward against gravity, through the outer race groove to a line at approximately a 45° angle from the opposite edge of the retainer (see Figure 3). There also is evidence of a fine film of oil on the bearing outer race below the contact zone as well as a thicker film on the lower outer race land. It is apparent that oil and some debris was pumped against gravity in a direction from the lube nut side of the bearing through the bearing outer race and out to the shallow land of the outer race (see Figure 3).

Test #3 was conducted using the same bearing as in the initial two tests. The retainer was replaced with another retainer which was soxhlet extracted in freon to remove residual oil and was then vacuum baked as stated previously. The bearing metal parts were then cleaned as prior to test #2. The bearing metal parts

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were dipped into a 50-1 mixture of freon and oil. The bearing metal parts and dry retainer were then assembled and replaced into the life test fixture.

Test #3 covered a period of approximately eight hours. The test was characterized by anomalies similar to that of test #2 and somewhat similar to some data from Skylab CMG #2. During the early hours of the run, while there was oil for adequate lubrication, there were no sudden changes in speed, current, or vibration level. During the middle hours of the test, variations in speed, current and vibration level occurred, but the changes in vibration level in most cases did not occur simultaneously with the speed and current changes. There were current and speed plateaus as in the data received from Skylab CMG #2, as well as ramps and recoveries. During the initial hours of the test there was no change in the bearing temperature of 73.5°F. The initial six hour period was characterized by plateaus and ripples in the current and speed traces of 10-30 ma and 60-80 RPM, although there are some instances of 50 ma current and 120 RPM speed changes.

The last two hours are characterized by more severe current changes of up to 200 ma. and 800 RPM changes in speed. There were three severe speed decreases to approximately 1000 RPM with current levels reaching 700 to 1150 ma. In each case the recovery in speed was complete. The test then was stopped overnight.

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The half hour run, during the following day, was similar to the last hour and a half of the previous run. However, audible retainer noise was noted during this period, and the resultant increases in vibration level were marked on the trace. The audible noise was consistent with speed and current fluctuations.

Area A occurred approximately one hour and twenty-five minutes into the test and is indicative of the initial hour and forty-five minutes of the run (see Figure 4). The current is 495 ma up 5 ma from the initial steady state reading. The speed is 9004 down from the initial 9036 RPM. The vibration level on the minimally lubricated (test bearing) bearing was steady at 60-80 mv. During this period of time, the lube end was sufficient to separate the balls and races and there were no anomalies present.

Area B (Figure 5) on the trace occurred approximately two hours and thirty minutes into the test and includes to anomalies. At the beginning of this period, the bearing indicated the first "gliche". The speed which was stable at 9000-9008 RPM drifted down to 8992 RPM and then dropped to 8947 RPM (point 1), while the current correspondingly increased from 495 ma to 515 ma. The speed then recovered slightly to about 8966 RPM then dropped to 8940 RPM with corresponding changes in current. There then occurred momentarily a complete recovery to 9022 RPM (point 2) and 480 ma.

14.3 (28.6) RPM

1502

1/2 SPEED

0.495 ma

Adv Current

100 ma

73.5°F

Temperature

100 sec



FIGURE 4

70 mV

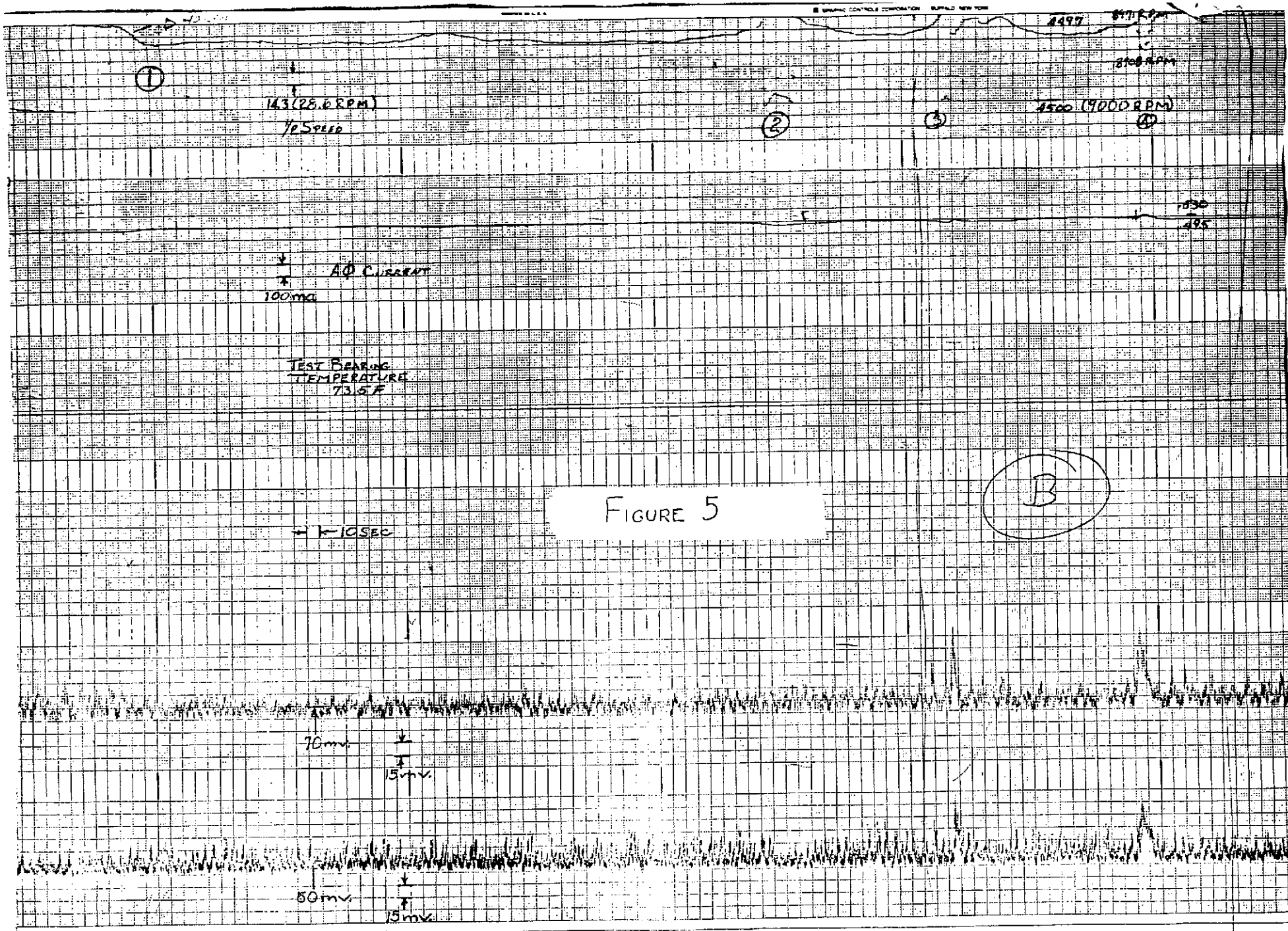
Vibration level

16 mV

50 mV

Vibration level

16 mV



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Points 3 and 4 were the first accuracies of simultaneous speed, current and vibration anomalies. There was no apparent change in test bearing temperature. These anomalies both cover a ten second period. Point 4 indicated a speed drop of 63 RPM from 8979 to 8908 RPM then a recovery to 8990 RPM. The current rose from 505 ma to 530 ma then decayed to 495 ma. These anomalies were the first indication of any significant bearing torque changes.

Area C - (see Figure 6) on the trace began about fifteen minutes after area B or about 2 hours and forty minutes into the test. There was a sharp sustained increase in vibration level from an average of 70 mv to an average of 100 mv. There are vibration peaks to 260 mv. (100 mv/g.) During this period there are examples of "glitches" in both the speed and current traces. The current ranged between 470 ma to 550 ma and the speed range varied between 9058 RPM and 8768 RPM. These anomalies recurred for approximately 30 minutes.

This period began abruptly with the sharp vibration level increase. The speed dropped from 7994 RPM to 8931 and then to 8900 RPM (point 1) with a corresponding increase in current. There was also a small temperature increase of approximately $0.2^{\circ} - 0.3^{\circ}\text{F}$ over a five minute period attributed to the initial torque increase of this period. Although the magnitude of the changes are different, as well as the time periods, the series

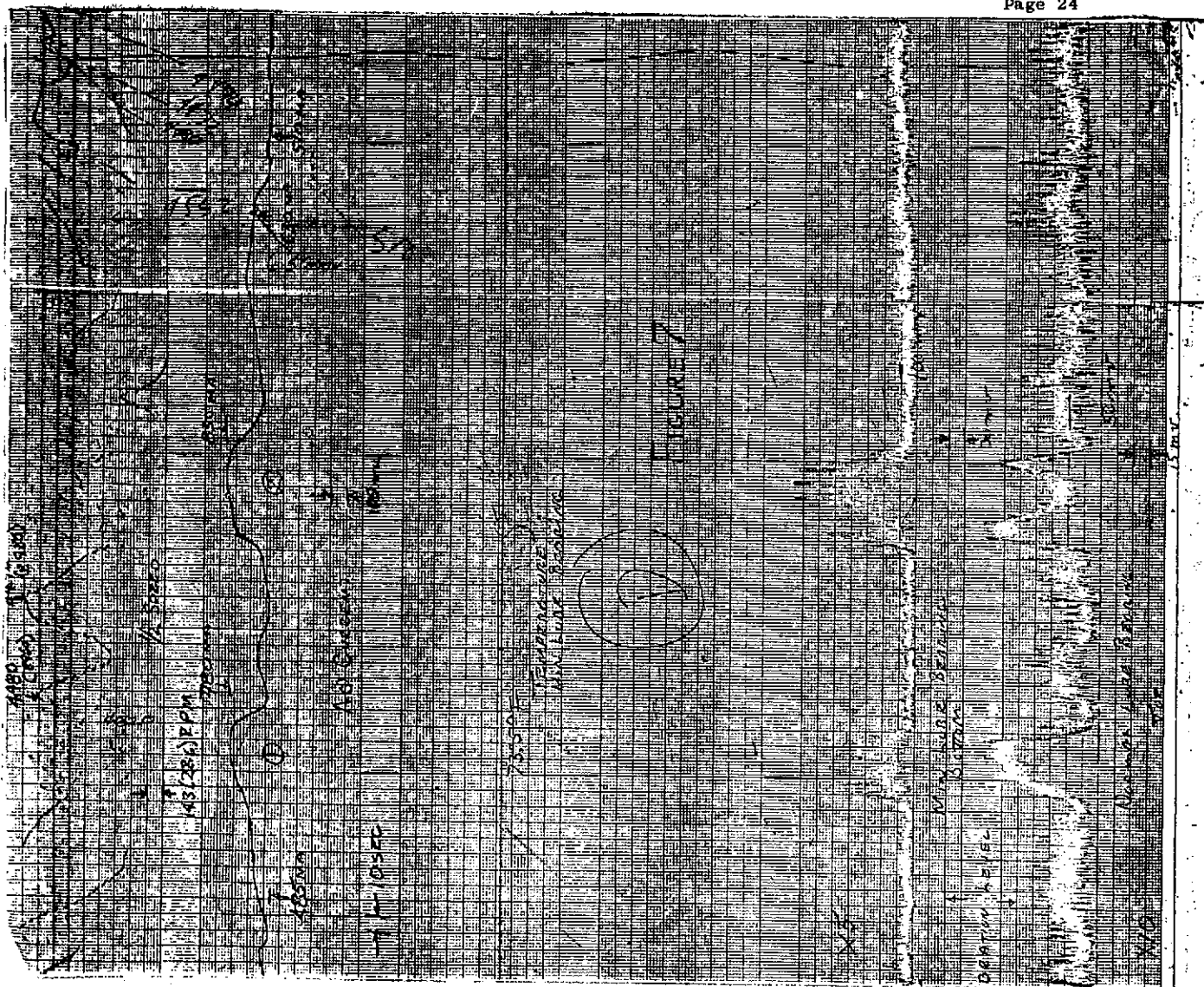
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of "glices: in area C are somewhat similar to the "glices" of DOY 323:16:45 to 323:18:15 (discussed later in this report).

Approximately 3-1/2 hours into the test, area D (see Figure 7) exhibited the first series of "failures" and "recoveries". The current level during the first "failure" reached 780 ma from 485 ma. The speed decreases during this period were too rapid to be recorded. The increase in vibration level was probably due to resonances in the system at lower speeds and did not precipitate the torque increases. The second "failure" was even more severe with the current level reaching 830 ma. In both cases there was a "recovery". The recovery from the first "failure" was back to 500 ma and approximately 8960 RPM. Then follows a milder speed decrease to 4425 counts or 8850 RPM from 8960 RPM, point 2. The second recovery was not as complete and was followed by another speed decrease which was not recorded. At the end of period D, the speed and current "glices" began to smooth out and were relatively stable for approximately the next two hours and fifteen minutes.

The "failures" and "recoveries" in this area were similar to those at the end of test #2 and were typical of the failure/recovery modes of bearings on the verge of failure.



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Area E (see Figure 8) occurred approximately four hours and forty minutes into the test and was indicative of the general recovery of the test bearing. This recovery period spanned approximately two and one-half hours and was punctuated by small "glices" of similar magnitude and duration as in area E. These "glices" were on the order of 20-55 RPM and 5-20 ma. The duration was usually 2-3 seconds. There were no temperature changes and during this recovery period the vibration level of the test bearing returned to previous levels of approximately 30-100 mv.

After approximately six hours into the test, the "glices" became more frequent and for the next hour increased in magnitude and duration.

Area F occurred approximately seven hours into the test (see Figure 9) and followed a series of "glices" of lesser magnitude and duration. The speed decreased from about 8950 RPM to 2148 RPM while the current increased from 520 ma to 1180 ma. During this period speed readings were impossible to determine from the trace and were read from the counter and written onto the trace. This "gliche" lasted about six minutes after which the speed and current became fairly stable and the vibration level dropped to normal levels and continued so for another half hour. The test was then shut down until the following day.

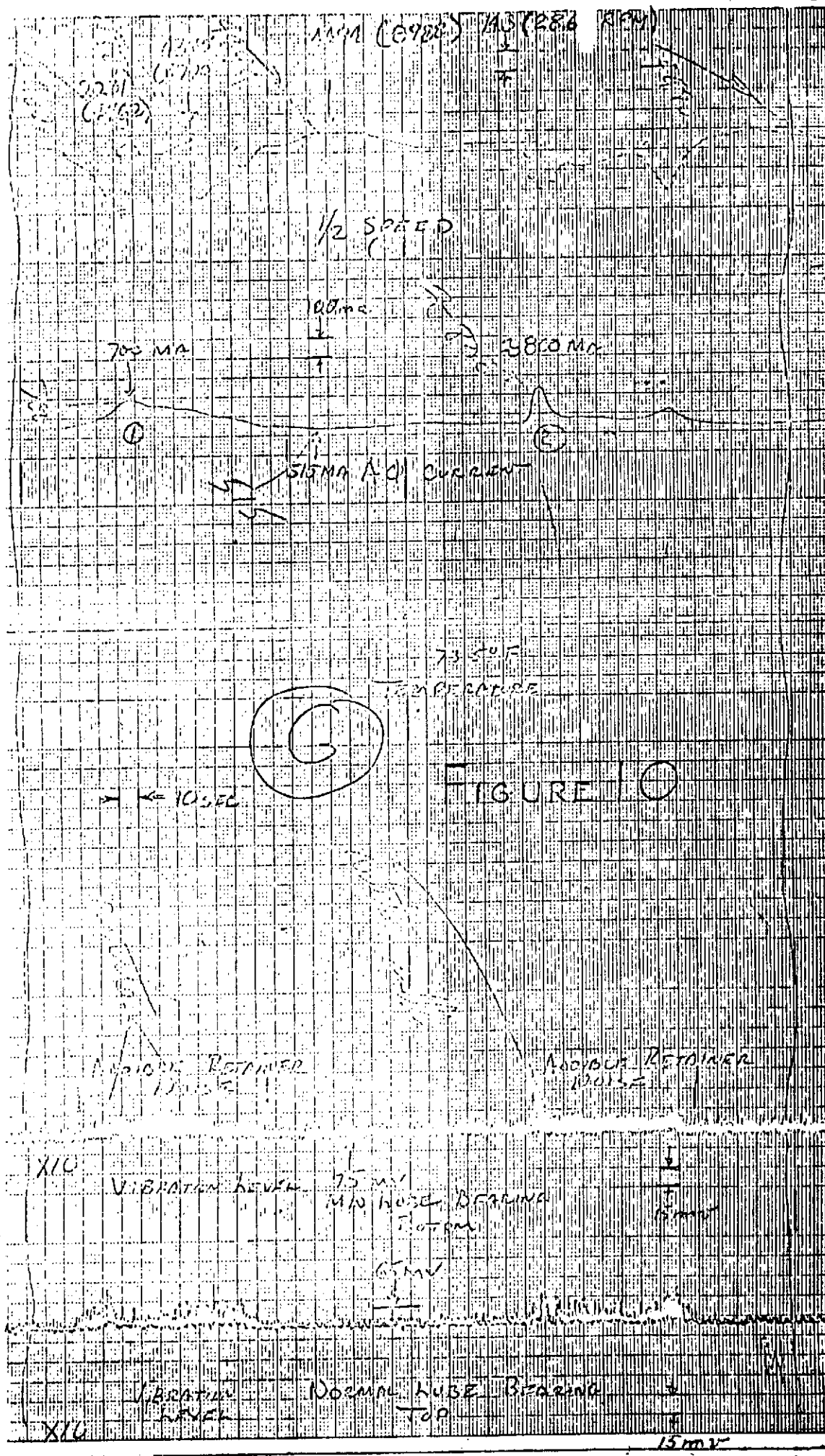
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Area G (see Figure 10) occurred after about twenty minutes into the next day's run and about ten minutes prior to termination of the test. Interesting points (1) and (2) on this trace were two of several instances of audible retainer noise (instability) corresponding with significant speed and current "glices". At point (1) the speed dropped from approximately 8970 RPM to 4462 RPM. The current change was from 540 ma to 800 ma. At point (2) the speed reading was not recorded while the current peaked to 800 ma.

The bearing was then removed from the life test fixture and disassembled for examination. Visual examination of the bearing metal parts and retainer indicated greater wear than generated after test #2. There was a similar pattern of debris and oil being pumped upward in a direction from the pressure side land of the outer race thru to the shallow land of the outer race. There was a greater amount of oil on the pressure side land than in test #2, with much of the oil from the lube nut migrating downward and away from the race. However, there was oil visible on the upper area of the outer race which had been pumped into the race.

The bearing metal parts showed additional wear, more so than in test #2, with evidence of some metal to metal contact between the balls and both inner and outer races. There was some minor spalling in the contact areas of each race. The balls also showed some minor spalling.



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The bearing inner race land did not indicate any polishing by the retainer pads nor do the pads show any wear. This indicated that the 50-1 solution provided a film sufficient for lubrication in this area.

There was a noticeable increase in the film of oil on the outer raceland, supplied by the lube nut. The majority of this oil migrated downward away from the bearing race. There was, however, a film of oil extending to the race edge and into the race. While no quantitative measurement of the thickness of this film or the quantity of oil could be made, it was by visual comparison at least several times the film left by the 50-1 mixture.

The retainer ball pockets were worn on the lead and lag segments to a greater extent than in test #2, and there was greater burnishing on the pocket sides. This indicated that this retainer was unstable for greater periods of time, and accounts for the correlation between audible retainer instability and the jogs in the vibration, current, and speed traces.

The inertia of the life test fixture shaft assembly is $.72 \text{ oz-in-sec}^2$ or $.045 \text{ in-lb-sec}^2$ (ref. MT-13,936), while the inertia of the IGRA is approximately 29.9 in-lb-sec^2 (ref. 2120252-9). The ratio of the inertia is $29.9/.045 = 650$. Therefore, assuming similar bearing drag torques for both the test bearing and the

bearings in the Skylab ATM CMG's, the effects of a change in torque in a Skylab CMG bearing would occur approximately 650 times slower than the speed and current reactions for the same torque change in the test bearings assuming that the speed of both is in the linear portion of the torque/speed curve (8700-9100 RPM (ref. MT-15,800 and Figure 12).

The data from Skylab DOY 323:16 - 323:19 shows a fairly typical "gliche" (see Figure 11). According to data received, the speed change during this period was from 8911 RPM to 8870 RPM, a change of 41 RPM. The torque change was approximately 1.2 oz-in from the above curve (see Figure 12). The speed change occurs over a period of approximately 20 minutes or 1200 seconds. A similar torque change in the life test fixture would occur within $1200/650 \approx 2$ seconds. The "gliche" noted in area B of test #3 (Figure 4, point 4) starts with a drop in speed from 8971 to 8908, a drop of 63 RPM and a torque change of 1.1 oz-in in a period of four or five seconds. Similarly, the plateau which began at point (1) of area C in test #3 began with a speed drop from 8994 RPM to 8931 RPM (63 RPM) and occurred in six seconds.

During DOY 349:11:15 - 349:49:14:15 (see Figure 13) the speed drops from 8910 to 8850 RPM in about 31 minutes, a torque increase of approximately 1.6 oz-in. This would have corresponded to 100 RPM changes within 3 seconds for the test bearings. Instances of changes of this magnitude and rapidity occurred during the last two hours of the test. One such is illustrated (see Figure 10) in area G during the second instance of audible noise.

REP-0540... 000003 VIEWID-HSP1 SEGS-0002 TO 0002 OF 0003
 SHS TRANSFER BUS CURRENTS

000003 VIEWID-HSP1 SEGS-0002 TO 0002 OF 0003
 SHS TRANSFER BUS CURRENTS

XS:
 EV:

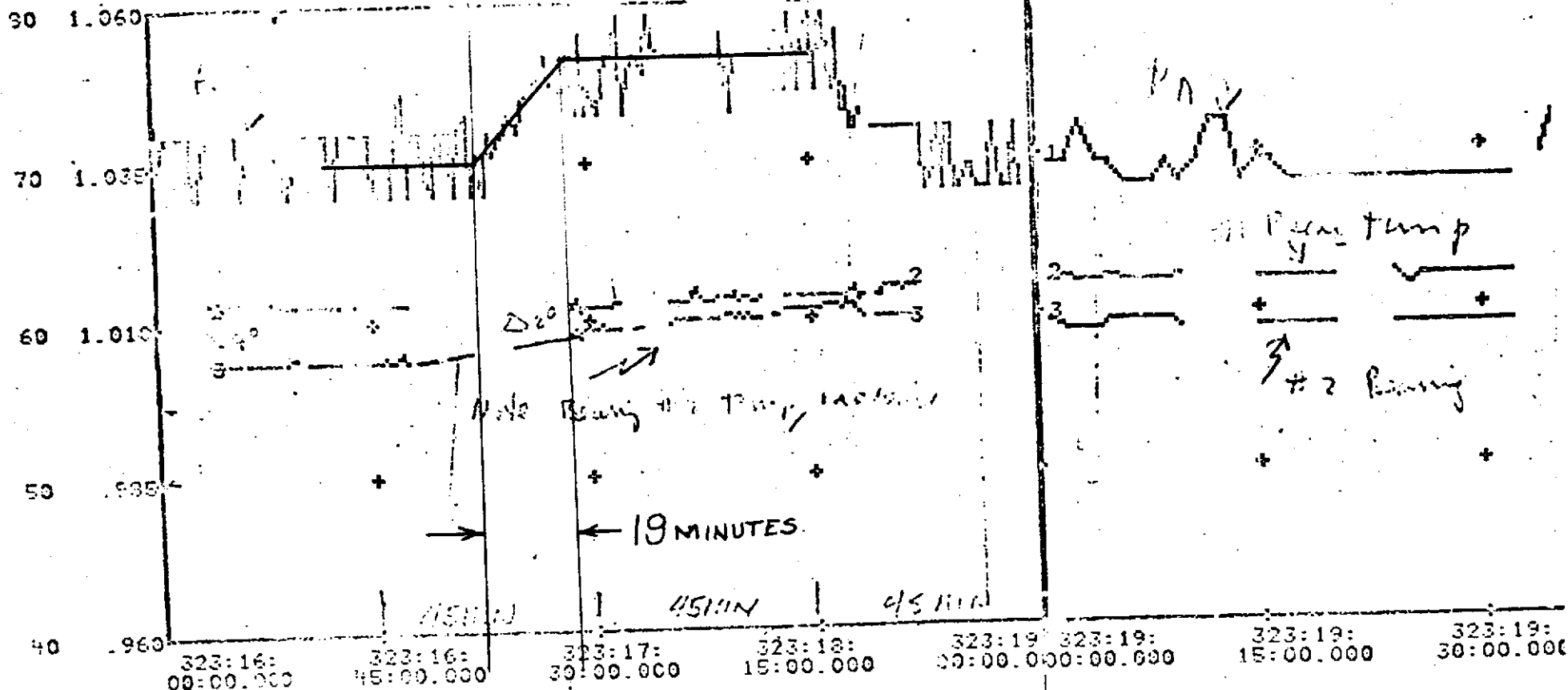
HHL A AMP
 00497-702 =1(R)

BRG 1 T F
 00008-702 =2(L)

BRG 2 T F
 00009-702 =3(L)

BRG 1 T F
 00008-702 =2(L)

BRG 2 T F
 00009-702 =3(L)



HCB

DAMP

FIGURE 11

415

LS1E2409

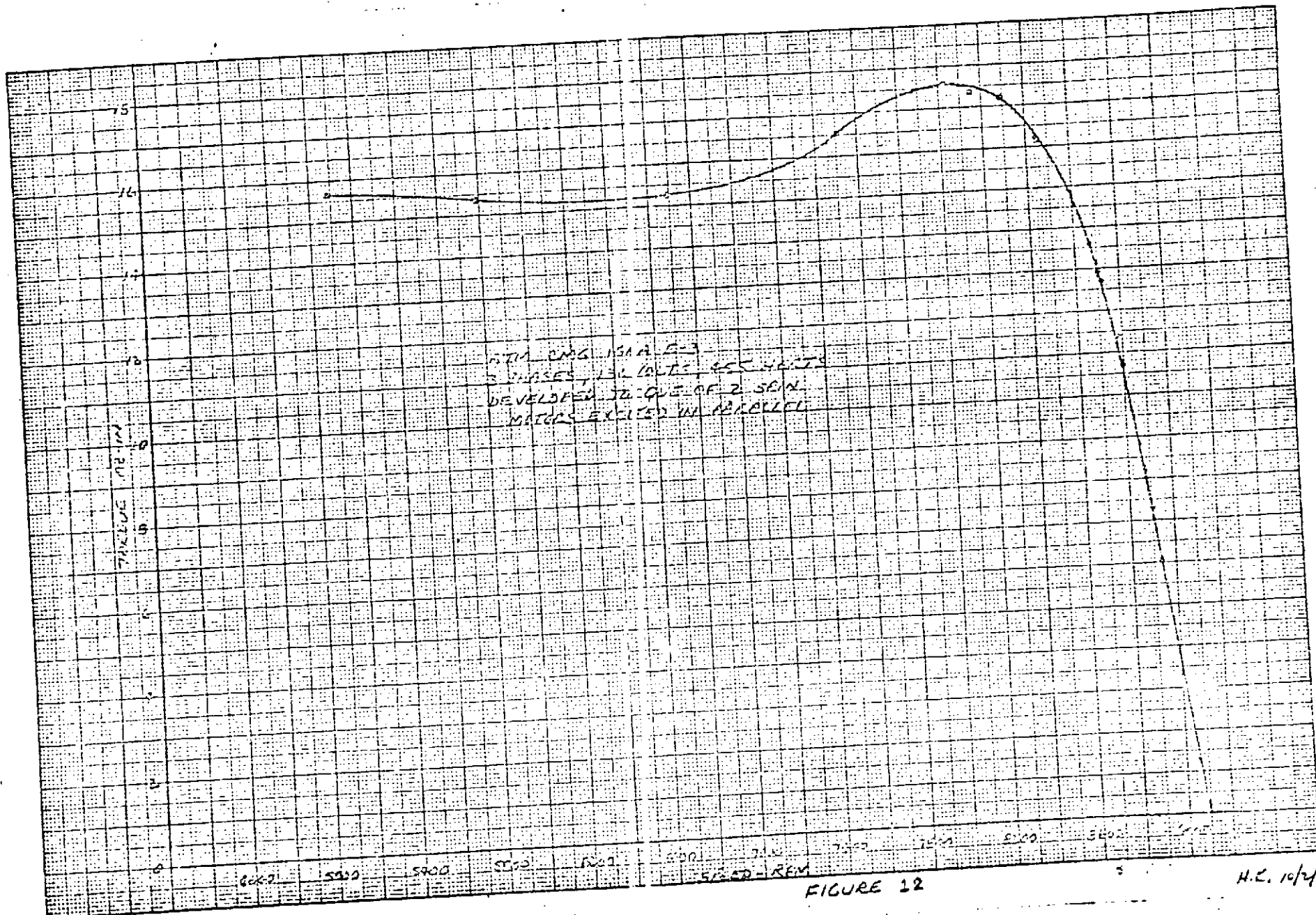


FIGURE 12

H.C. 10/4/72

FIGURE 12

6

0002 TO 0002 OF 0003
REPORT FOR BRASHER (SPR-N)

NUM-0101 VIEWID-MSP4 SEGS-0002 TO 0002 OF 0003
REPORT FOR BRASHER (SPR-N)

XS: GMT
EV: EVT0100L

XS: GMT
EV: EVT0100L

GMT

M0487-702 =1(L) C0009-702 =3(R)

WL) C0008-702 =2(R) C0009-702 =3(R)

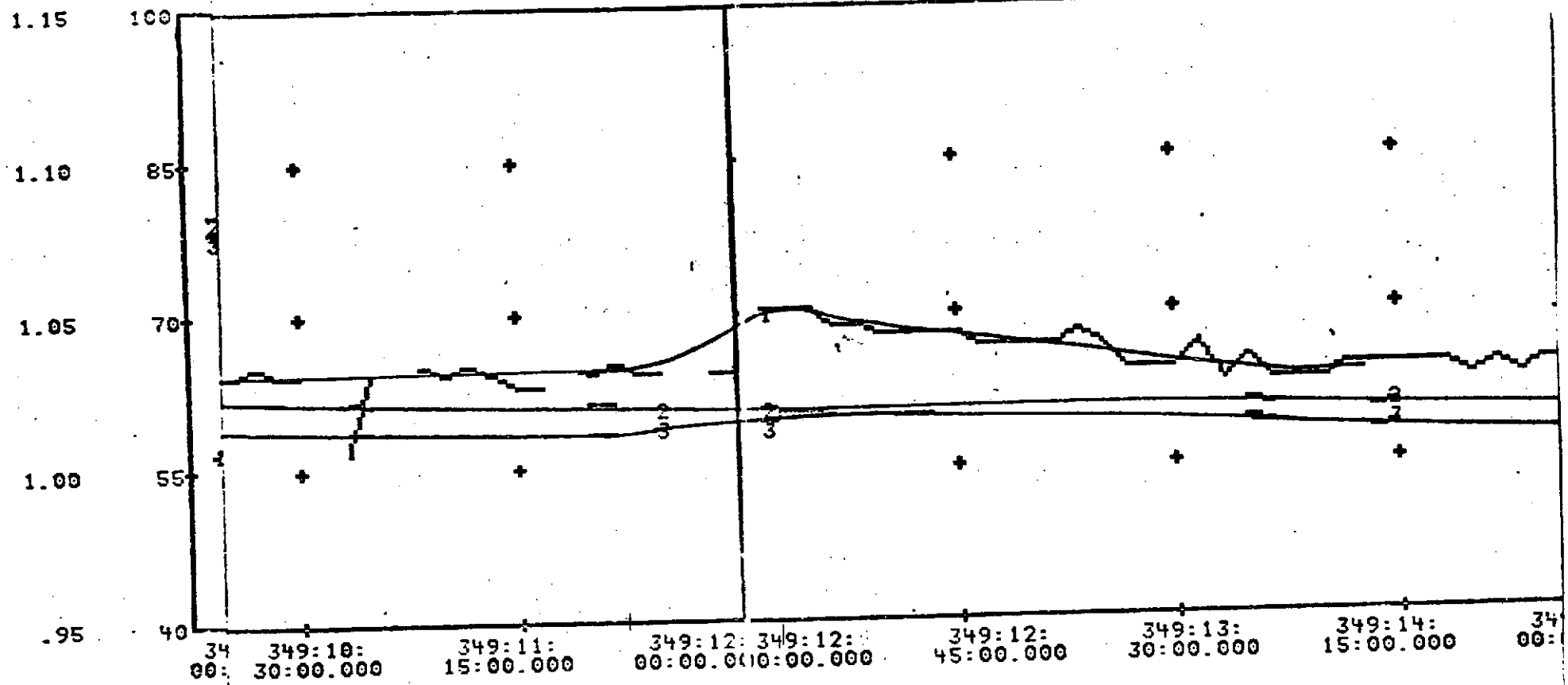


FIGURE 13

LSI2 WOS

RLP4-349:09:14

5.0 CONCLUSIONS

In the three tests, the primary cause for "failure" was the buildup of debris and residues from localized overheating in the unlubricated ball bearing retainer pockets due to ball rub.

The location of debris, residues and small quantities of oil in the outer race indicates that some oil is pumped or migrates against 1 g to the bearing contact area. However, with the standard retainer used in this test, the deposition of replacement oil from the lube nut was onto the outer race land and did not migrate against 1 "g" adequately to relubricate the test bearing.

The condition of the test retainers scalloped pads after each test indicates that the 50-1 mixture of freon to oil leaves the minimum oil necessary to adequately lubricate the inner race land/scallop pad interface.

In the tests, audible retainer noise did not always occur in correlation with, or cause a change in torque level (wheel speed). However, there are instances in both tests #2 and #3 where audible retainer instability is a cause for the torque level change in the test bearing. In the tests, instances of audible retainer noise did not produce "gliche" plateaus of fairly constant speed and current. They generally produced current spikes and irregular speeds during the audibly unstable periods. However, in the actual CMG's with their larger inertias these changes would probably have smoothed out.

Issue: Original

MT-15,786

Date: 6 January 1975

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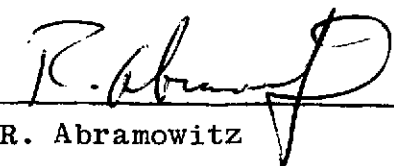
There were external factors occurring to the Skylab CMG's such as thermal cycling, gravity gradient momentum desaturations, etc. which could not be duplicated in these tests. Therefore direct correlations between the "glices" seen in the telemetered data from Skylab ATM-CMG's 1 and 2 and the data taken during these simulation tests are difficult. There appeared to be some similarities in the data, particularly during the first half of test #3, where torque level changes were of similar magnitudes, although reactions in speed were not quite as fast as the 650/1 ratio of inertias (ATM-CMG wheel/test fixture shaft). The similarity in some of the test data to some of the telemetered data from Skylab does support to some extent the theory that minimal lubrication was the primary cause for the anomalies noted.

TO: Engineering File MT 15,775
From: A. Carriero

Issue: Original
Date: October 24, 1974

LIFE TEST OF NORMALLY LUBRICATED ATM-CMG
BEARINGS WITHOUT LUBRICATION REPLENISHMENT

Approved by:


R. Abramowitz

Prepared by:


A. Carriero

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY

Issue: Original
Date: October 24, 1974

MT 15,775
Page 1

1.0 PURPOSE

The purpose of this report is to document the results of a bearing life test in which ATM CMG bearings were run in a life test fixture without benefit of a lubrication replacement system. The test was designed to simulate a possible condition on Skylab in space where little or no oil is replaced to lubricate the bearing.

2.0 BACKGROUND

One area of investigation into the possible causes for the failure of Skylab ATM CMG #1 (CMG S/N 5) and the anomalies in the speed current and temperature data telemetered from Skylab ATM CMG #2 (S/N 6) is inadequate relubrication of the spin axis bearings. Therefore, the purpose of the test conducted in this report is to determine the period of time bearings would operate satisfactorily without oil replenishment.

3.0 DISCUSSION

It is difficult to simulate space conditions on earth. However, some approximations to space conditions can be made. To simulate the weightlessness in space, the bearings were run in a life test fixture. The life test contains only a shaft of negligible weight compared with the 150# total weight of the CMG wheel. The bearing axial or preload was set at 40# as in the Skylab CMGs.

The life test fixture cavity was pumped down to a pressure of approximately 5×10^{-3} Torr.

While the pressure in space will be in the area of 10^{-9}

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10^{-11} torr the labyrinth seal and other factors should keep the pressure in the ATM CMG at or below the vapor pressure of KG80 oil which is 10^{-7} at 100°F and approximately 10^{-8} at 80°F . However, the vapor pressure of the oil is extrapolated to be 10^{-9} at 100° after evaporative loss of 5% by weight of the light ends of the oil and 5×10^{-10} at 80°F (ref 1). Labyrinth seal and molecular flow analysis estimates the loss of lubricant from each bearing to be approximately 1.6×10^{-6} gm/hour if the temperature of the oil is at 100°F and the surrounding enclosure is at 60°F , the low point of the ATM CMG temperature range in Skylab. The escape of oil vapor thru the labyrinth seal is estimated to be less than 1.0×10^{-6} gm/hr.

The ATM CMG bearings in space will contain in the order of 75×10^{-3} gm of oil impregnated into the retainer and an additional 50×10^{-3} gm of oil added to the races prior to assembly into the CMG. There is an additional supply of approximately 40×10^{-6} gm of oil each hour the CMG is run on earth. If the worst case is considered in which the only oil available to the bearing is the 50 mg added prior to assembly and 1/3 of the oil impregnated into the retainer, there is approximately 75 mg available to lubricate each bearing. At the above evaporation rate, 5% of this oil or 3.75 mg will take over 2000 hours to evaporate. At the point where 5% of the available oil has evaporated, the vapor pressure of the oil has decreased by 10^{-2} torr and the evaporation rate will decrease to $.016 \times 10^{-6}$ gm/hr.

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Therefore the effect of lubricant evaporation is relatively minor with respect to bearing life.

Pumping down the life test fixture serves two purposes; it eliminates windage effects as in the ATM CMG in space, and removes oxygen from the bearing environment. This will deprive the lubricant and bearing metal interface of oxygen resupply which may have some effect on bearing life in space.

In order to simulate lubricant creep and possible loss of lubricant due to earth test in the spin axis vertical position, this test was run with the spin axis vertical such that gravity as well as thermal gradient would cause lubricant to creep from the race contact areas.

To simulate the bearing speed in space, the life test fixture was also run at 9100 rpm.

The two bearings selected for this test both have had a history of use both in ATM CMG wheels and in various life tests during the past several years. They were selected because both had accumulated several hundred hours of run with a lubrication replacement system and had the quantities and dispersion of oil within the bearing metal components and retainer similar to that of the bearings used in the Skylab ATM CMGs. Therefore, the bearing metal wear and ball pocket rub was also similar to those bearings used in the three Skylab AMT CMGs.

In late 1972 a somewhat similar test was run at Marshall

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Space Flight Center utilizing a minimally lubricated bearing. This bearing contained 21.4 mg of oil in the retainer and an additional 6 mg of oil on the races. This is in comparison with the normal 50-80 mg of oil in the retainer and 50 mg in the races prior to assembly into an ATM-CMG. This test ran for 400 hours at 40# preload with the life test fixture on the spin axis horizontal position and an additional 355 hours horizontal at 190# preload to simulate the load of an ATM CMG spin axis vertical situation on earth. Although the bearing torque increased after the 355 hour period, the bearing metal parts indicated no degradation.

4.0 RESULTS

The bearings in this test ran for a total of 1688 hours. Torque and speed measurements were taken and recorded each day. The initial torque measurements on April 24, 1974 was 1.29 oz inches. The bearing then ran for 1600 hours at torque levels between 1.53 and 1.85 oz. in. The speed variations during this period were 54 rpm. Due to the utilization of test equipment for other concurrent testing, continuous vibration level bearing temperature and motor current readings were not made. However, random measurements of these parameters were made during the entire period of the test with no significant variations from steady state running values.

On July 8, 1974 after running 1688 hours, the torque test indicated a bearing torque of 2.45 oz in up from 1.72 on July 5, 1974.

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The speed had also decreased from 9088 rpm to 9065 rpm. This significant torque increase indicated that one or both bearings were beginning to fail and the test was terminated.

Both bearings were removed for visual examination. Bearing S/N SBB015 still contained significant amounts of oil on its metal parts and on the retainer surfaces. The bearing contact areas and ball wear pattern were visually not significantly deteriorated from when the test began. There were no significant buildup of pyrolyzed oil or wear metal residues on the ball pockets and no evidence of retainer instability.

Bearing S/N B19 the top bearing ran dry and was the bearing which exhibited the increase in torque. Visual inspection of the bearing indicated that the bearing retainer was completely dry. The ball pockets contained buildups of pyrolyzed oil in the lead and lag portions of the pockets. There was only minor burnishing of the pockets in these areas. There was no evidence of wear or debris from contact between the retainer scallop pads and the inner race lands.

The bearing inner and outer races contained pyrolyzed oil imbedded into the contact zones and pushed out to the periphery of the contact areas. There were minor amounts of free oil visible outside of the contact areas which either migrated from the contact area or had never migrated into contact area. The bearing balls are coated with pyrolyzed oil with some balls containing areas of

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pyrolyzed oil imbedded on the ball surface.

Since the test was terminated at the first instance of a significant torque change, there was no apparent metal damage, i.e. fretting or spalling, to the bearing metal parts. The torque increase was caused by the buildup of wax-like pyrolyzed oil residues in the race contact areas and on the balls.

5.0 CONCLUSIONS

Based on the results of this test and of the minimal lubrication test (ref 2), it appears that an ATM CMG bearing which is properly lubricated initially, will run without further lubrication, for at least 1500 hours in space conditions. The bearing which "failed" would have continued to perform its function on a CMG for an additional several hundred hours. However, variations in torque current, and speed would be noted.

The appearance of the "unfailed" bearing indicates that it would have run for a period in excess of 3000 hours. This 3000 hours may be closer to the average life to be expected. The minimal lubrication test (ref 2) bearing contained less than 20% of the normal quantity of oil present initially on an ATM CMG bearing. It ran for 755 hours approximately half of which was under loads five times that which would have been experienced in non-maneuver modes in space.

The failed bearing, S/N B19, had not reached a critical point where catastrophic failure was eminent. Although

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much of the oil was pyrolyzed and the effectiveness of the hydrodynamic film degraded, the friction between the balls and ball pockets had not reached the point where there is localized overheating of the phenolic in the ball pockets, and generation of resinous debris. Likewise there was no evidence of metal to metal contact.

In the ATM CMG operating temperature range of 60°-80°F and with a labyrinth seal, the evaporation rate of the KG-80 oil used to lubricate the bearings is minimal and is a minor factor affecting bearing life.


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"Bearing Verification Tests for the Bendix Apollo Telescope Mount Control Moment Gyro" Final Report
Bendix Research Laboratories Report No. 4398, April 1968, page 9.
2. Carriero, A., "Minimal Lubrication Tests ATM-CMG Bearing" Bendix Navigation and Control Division
Report NCER-73-06-008, January 1973.

To: Engineering File - MT-15,776
From: A. Carriero

Issue: Original
Date: October 24, 1974

ATM FAILURE ANALYSIS
OIL MIGRATION AGAINST IG

Prepared by: 

A. Carriero

Approved by: 

R. Abramowitz

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY

Issue: Original
Date: Oct. 24, 1974

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Page 1

1.0 PURPOSE

The purpose of this report is to document the results of a test to determine whether oil centrifuged onto an ATM-CMG bearing outer race land will flow against gravity into the race contact area. The tests were conducted with the CMG bearing spin axis vertical such that any oil migrating into the race contact area will have migrated against gravity.

2.0 BACKGROUND

The failure of CMG No. 5 (Skylab CMG 1) and the anomalies which occurred with CMG No. 6 (Skylab CMG No. 2) caused the review of all aspects of the ATM-CMG lubrication replacement system. One aspect reviewed is the migration of oil into the bearing outer race contact area from where it is deposited on the land near the race edge. MT-15,777 covers the design changes which will deposit the oil directly into the bearing race contact area.

Life tests (Ref. NCER 73-06-042) with the CMG spin axis horizontal have proved that adequate oil migrates into the bearing race contact area while running in this orientation. Life tests in excess of 30,000 hours were made in this orientation.

ATM-CMG qualification and preflight tests were also performed in the spin axis vertical position. However, the amount of test hours in this orientation was small as compared with the bulk of testing and run-in which was performed in the spin axis horizontal position.

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3.0 DISCUSSION

The tests were performed under the following conditions:

1. The bearing metal parts were cleaned and then dipped into a 50-1 mixture of freon and KG-80 oil to provide for a minimal hydrodynamic film.
2. The bearing phenolic separator was completely dry except for the ball pockets which were each "wetted" with approximately 1.5 mg of KG-80 oil, the approximate amount impregnated into a flight retainer pocket. This was accomplished by applying 10 drops of the 50-1 mixture to each pocket using a #20 hyperdermic needle. The purpose of lubricating the pockets was to prevent localized hot spots in the pocket contact areas.
3. The lube nut used contained oil tinted with a red dye and had been used successfully in previous tests (Ref. NCER-73-06-042).
4. The test bearing was run in a life test fixture with the bearing spin axis vertical. The test bearing was the bottom bearing with the lube nut situated below the bearing. Therefore, the flow of any oil to the race contact area was against gravity.

Since the first test bearing failed after approximately 140 hours, but during an overnight period, a second-test bearing was also run. In both tests the bearing was removed from the life test fixture, visually examined, photographed after the initial 50 hours of run. The second bearing was also removed and photographed after 100 hours.

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Both units were examined and photographed after failure at approximately 160 hours. After each examination, the bearing was reassembled without additional oil. The bearing race orientation was always such that gravity was acting against migration of oil to the race contact area.

4.0 RESULTS

Examination of the oil on the bearings surfaces after 50 hours of the first test revealed a thick ring of red oil at the bottom of the race land where it had collected after being centrifuged from the bearing retainer onto the land. See photo 1 and Figure 1. A thin film of oil was visible on the race land from this point up to the race corner. The bearing outer race contact area and balls appear "wet", but the film is too thin to detect any "redness". The film is in excess of the film left from the 50-1 freon-KG-80 dip.

The bearing retainer oil groove did not contain any excess oil. Therefore, all the oil slung off the lube nut was centrifuged thru the oil holes to the race land.

The results of visual examination after 50 hours of the second test bearing were similar to that of the first test. No visual examination was made on the first unit after 100 hours. Visual examination of the metal parts of the second bearing after 100 hours of run revealed the following: A ring of red dyed oil approximately 1/8" wide at the bottom of the outer race land (see Figure 2 and Photo 2). A thin film of oil was visible from the ring upward to the race corner. There was a thinner film of oil in the upper area of the race groove and some oil present in the race contact

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area as noted by light defraction in some areas. There is a similar very thin film on the balls and the inner race contact area. None of this film was thick enough to visibly fluoresce under untra-violet light. Some retainer pocket wear products and pyrolyzed oil were visible at the periphery of the race contact areas. There was also pyrolyzed oil and metal wear products deposited on the lead and lag portions of the ball pockets.

After reassembly of the bearing components, the second bearing ran an additional sixty hours to failure.

Examination of the first bearing after the failure at 140 hours revealed a ring of red dyed oil approximately 3/16" wide at the bottom groove of the race land (see Figure 3). There was also a film of oil extending upward from this ring up to the race corner. In the upper part of the bearing outer race groove, there was free oil visible but covered with debris and wear products of the failure. The bearing contact area contained residues and debris, but no free oil was noted by visual means. Use of ultra-violet light to determine the presence of KG-80 oil in the vicinity of the race resulted in the following observations: the thickness of oil film among the debris in the upper area of the race was sufficient to fluoresce faintly. The film of oil in the contact area of the outer race, and the balance of the race upward from the race groove which had been exposed only to the 50-1 freon to KG-80 oil mixture, did not contain enough film thickness to fluoresce.

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There was no visible fluorescence from oil present on the inner race. There was also no visible fluorescence from the retainer ball pockets or from the retainer groove overlapping the lube nut. The 3/16" wide ring of oil was readily fluorescent under ultra-violet light.

Photos 3 and 4 show the oil ring at the bottom of the outer race. Photo 4 also shows the wear products of the failure, and the discolored oil and residues in the race contact area. The brighter area above the discolored wear pattern is reflected light from the film of oil present but obscured by wear debris. Photo 5 is another of this area.

Figure 6 shows the wear and debris buildup on the ball pockets of the retainer. The elliptical contact area in the lead and lag portions of the pockets was burnished but contains no pyrolyzed oil. The debris and pyrolyzed oil was deposited or pushed to the periphery of the contact areas. The rub pattern on the ball pockets indicates minor axial retainer instability during this test.

5.0 CONCLUSIONS

The results of the two tests performed indicate that on earth some of the oil centrifuged onto the race land in the proximity of the race corner does "wet" against gravity into the bearing outer race contact area. However, most

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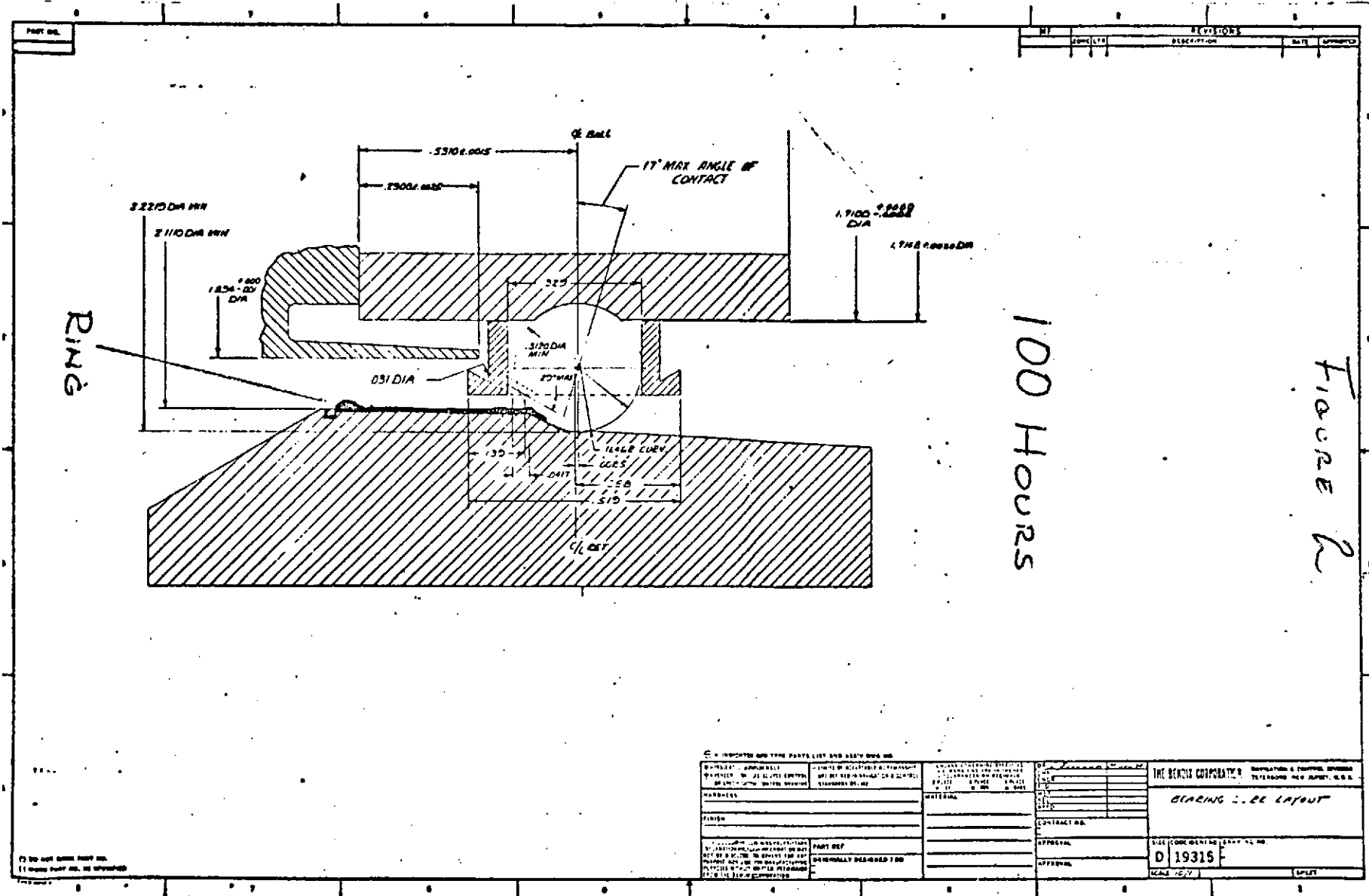
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of the oil drains downward away from the outer race groove corner and is useless for lubrication of the bearing.

The amount and rate of oil migrating into the outer race under these conditions is insufficient to sustain the hydrodynamic film in a bearing which is marginally lubricated.

The location of debris on the outer race indicates pumping action between the retainer OD and outer race land ID such that there is some flow of debris and oil against gravity.

There was insufficient quantities of red dyed oil transferred from the outer race to the inner race to be visibly detected on the inner race.



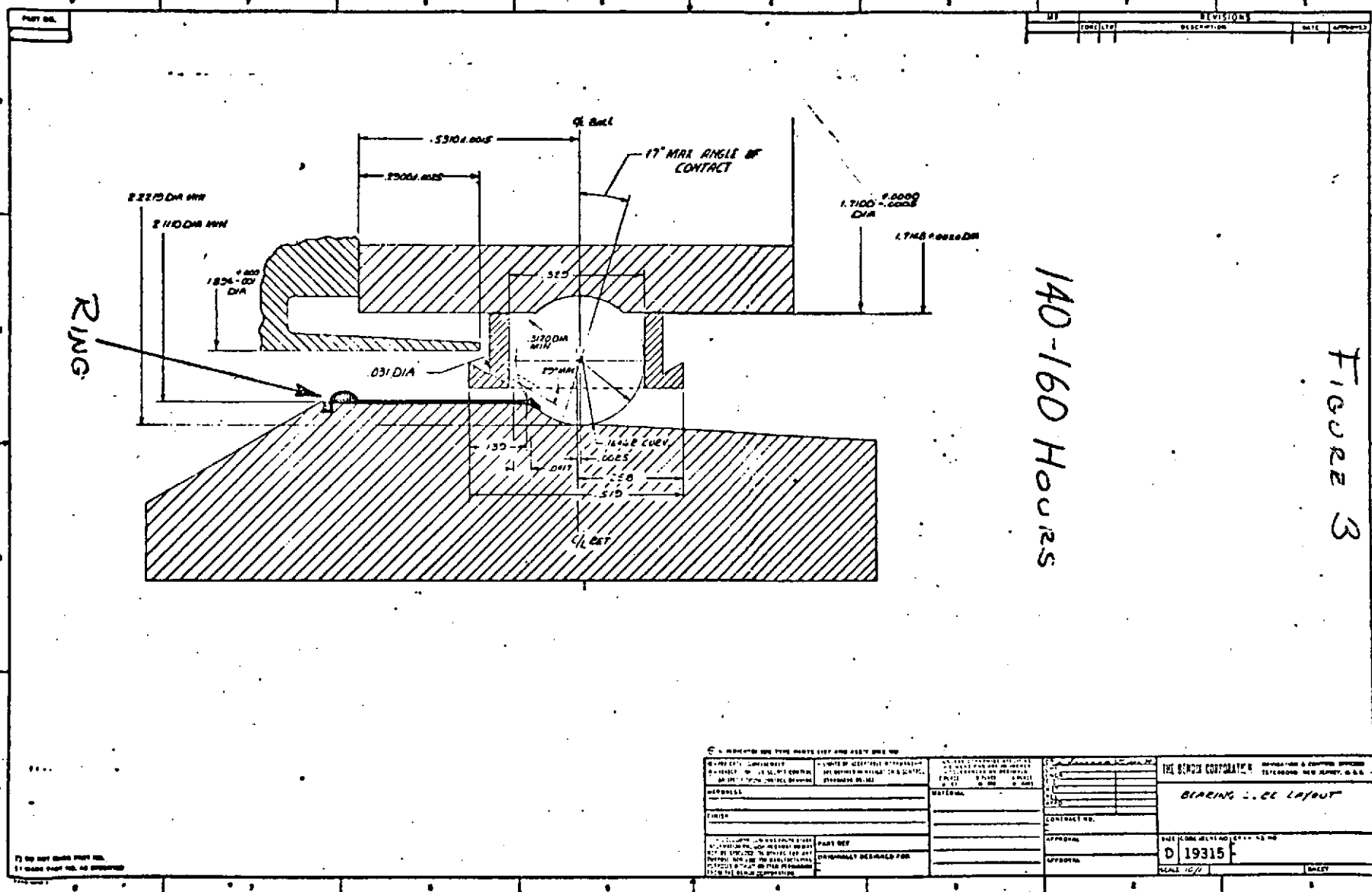




PHOTO 1

FAINT RED RING AT BOTTOM OF DISASSEMBLED
OUTER RACE 50 HOURS 1ST TEST

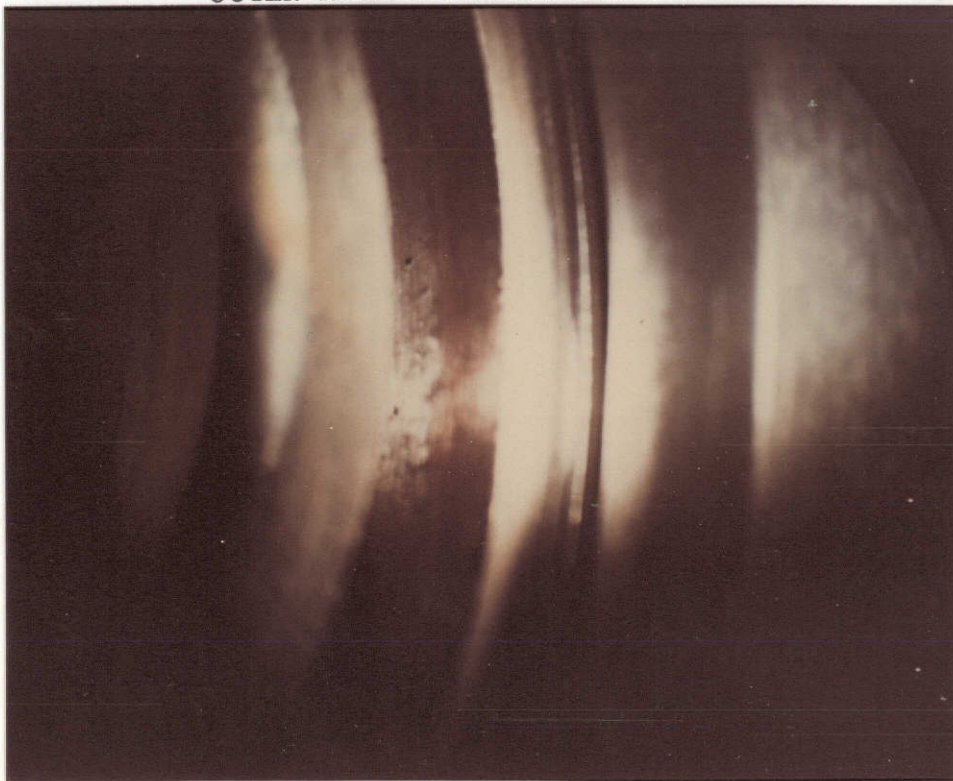


PHOTO 2

RED RING AT BOTTOM OF ASSEMBLED
OUTER RACE 100 HOURS TEST 2

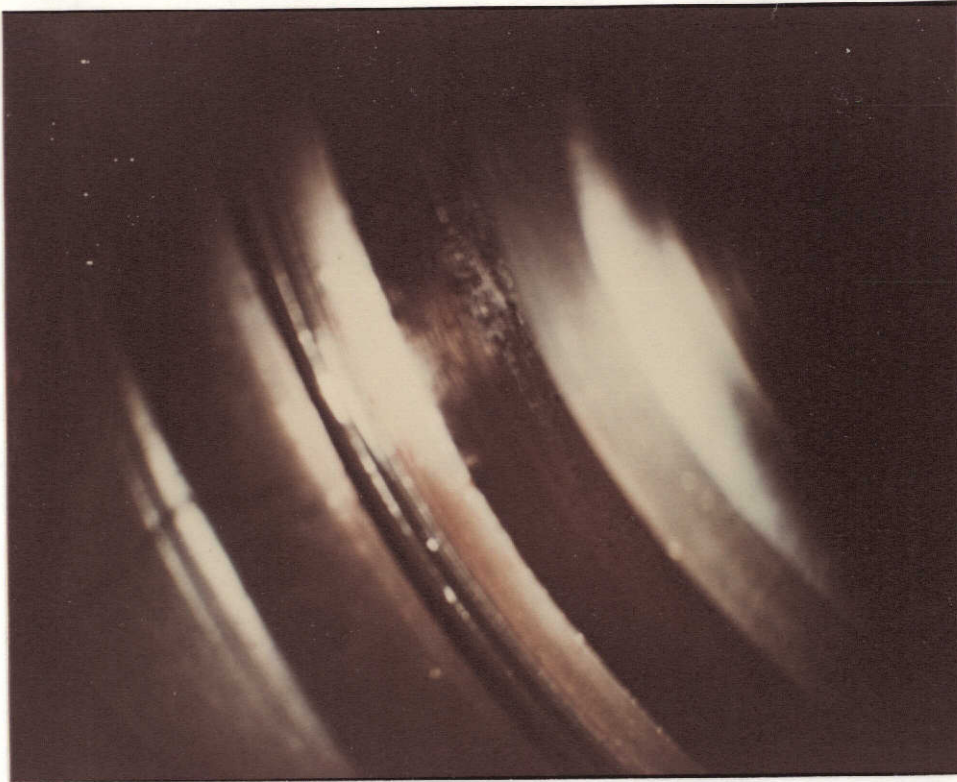


PHOTO 3

OIL RING AT BOTTOM OF ASSEMBLED
OUTER RACE 160 HOURS TEST 2



OIL RING AND WEAR DEBRIS
DISASSEMBLED OUTER RACE
160 HOURS TEST 2



PHOTO 5
OUTER RACE GROOVE
REFLECTED LIGHT
FROM OIL IN RACE GROOVE



PHOTO 6
RETAINER BALL POCKETS 160 HOURS

To: Engineering File MT-15,782
From: C. Gurrisi

Issue: Original
Date: 10 December 1974

HYDRODYNAMIC FILM TESTING
OF THE
ATM, CMG INNER GIMBAL
AND
ROTOR ASSEMBLY (IGRA)
UTILIZING 107-H ANGULAR CONTACT SPIN AXIS BEARINGS

Prepared by: C. Gurrisi
C. Gurrisi

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608

Issue: Original

Date: 10 December 1974

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PURPOSE

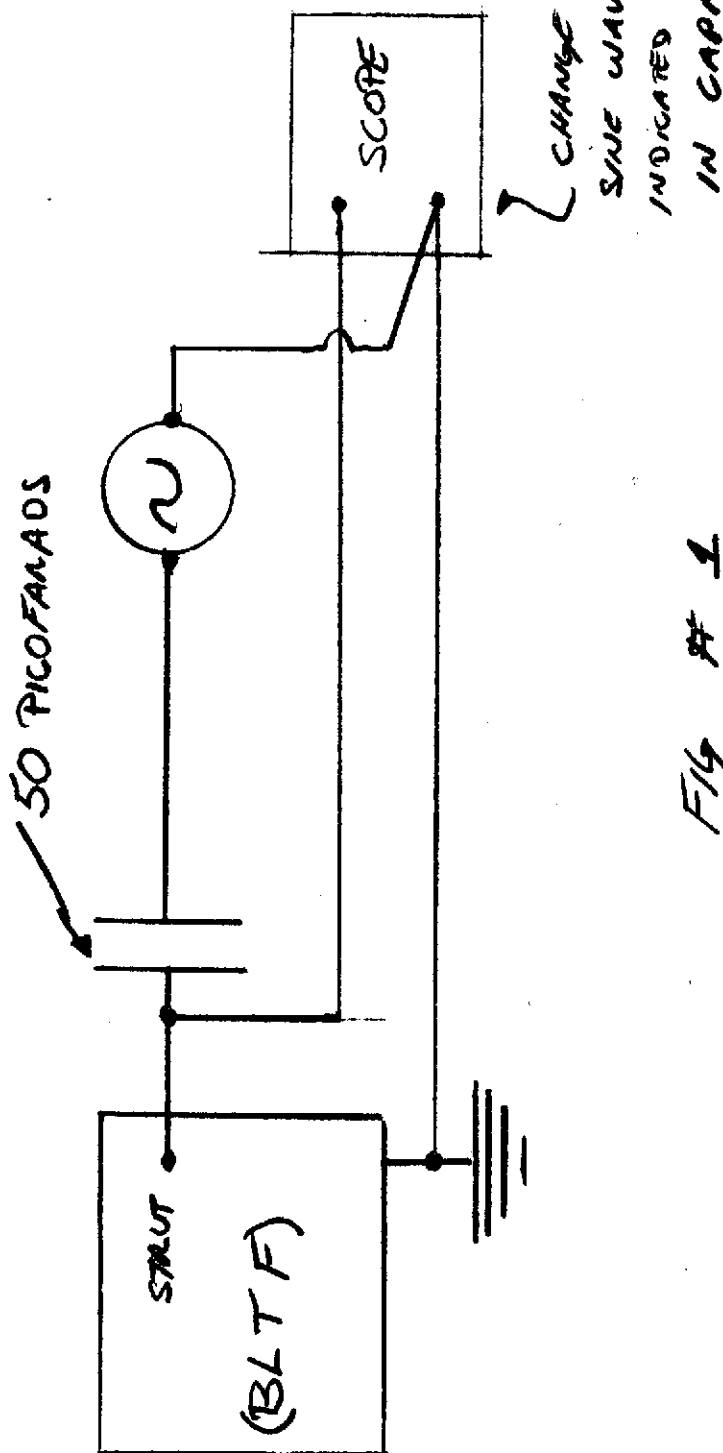
This test program was initiated to determine the wheel speed that develops a sufficient hydrodynamic film in the spin axis bearing as a function of temperature, cavity pressure, and lubricant quantity.

DISCUSSION

For this test program two ATM CMG Inner Gimbal and Rotor Assembly (IGRA) angular contact ball bearings were normally lubricated with KG-80 oil, torque tested for quality, and assembled into an IGRA Bearing Life Test Fixture (BLTF). This BLTF was modified for special capacitance measurements to determine the development of a hydrodynamic film in the spin axis bearings. The stiffening thru strut was electrically insulated from the bearing outer race (i.e. the case of the BLTF) by employment of lucite end caps. Any capacitance increase between the thru strut and case of the BLTF would indicate the presence of a hydrodynamic film within the ball bearings. The circuit used to monitor this capacitance is shown in Figure 1.

The first attempt at determining the wheel speed that develops a hydrodynamic film was performed on this BLTF at room temperature, and a cavity pressure of 5 microns. The capacitance measurement indicated a film was present before the wheel speed indicator was triggered. Therefore, the second test was performed at

HYDRO DYNAMIC FILM SCHEMATIC.



atmospheric pressure with the side covers removed so wheel speed could be detected by means of a strobe light. This set-up also produced unsatisfactory results because the strobe light introduced pulsating electrical noise into the capacitance measurement. The third attempt was also performed at atmospheric pressure with the side covers removed. However, the bearing lock nut was marked so that the wheel speed could be determined by counting the revolutions per unit time. The wheel speed of the BLTF was gradually increased until a lubricant film was indicated by the capacitance measuring electronics. This technique showed a hydrodynamic film was present in both bearings at speeds of 66-84 RPM.

Since speed detection at low speed presented a large problem in these initial tests, successful testing in thermal-vacuum conditions was doubtful. The BLTF was subjected to a temperature of 165°F and a vacuum of 10^{-7} Torr in the thermal vacuum chamber. Once thermal equilibrium was reached, the unit was slowly accelerated to operating speed. Throughout the accelerating period the capacitance electronics were saturated with noise, caused by long cabling to the thermal vacuum chamber. To obtain any satisfactory results in thermal vacuum environments a more sophisticated means of measuring speed and a special capacitance bridge will be needed. Since these items are considered special test equipment, not funded by the present contract, and the results of further testing was uncertain, it was decided in conjunction with NASA personnel to discontinue the task.

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CONCLUSIONS AND RECOMMENDATIONS

This test indicated that a hydrodynamic lubricant film is developed over a speed range of 66-84 RPM and is maintained throughout the run-up period to 9100 RPM. Since this hydrodynamic film was established at a very low speed, in comparison to the operational speed (9100 RPM) of the CMG, insufficient hydrodynamic film lubrication does not appear to have been a problem in the ATM CMG.

If any further hydrodynamic film testing is to be considered, a very accurate and low speed detection system will have to be developed, along with a sophisticated capacitance bridge.

To: Engineering File - MT-15,784
Date: A. Carriero

Issue: Original
Date: January 6, 1975

LIFE TESTS WITH
REDESIGNED ATM CMG RETAINER

Prepared by:


A. Carriero

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY

Issue: Original
Date: Jan. 6, 1975

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1.0 PURPOSE

The purpose of this report was to document the results of life tests utilizing ATM CMG bearings and redesigned bearing retainers. The bearing retainer was redesigned to more effectively capture and distribute oil to the bearing contact surfaces from the lubrication replacement system (lube nuts). The purpose of these tests was to evaluate the effectiveness of the design changes.

2.0 BACKGROUND

The primary mechanisms used to orient and maneuver the Skylab space laboratory were three Control Moment Gyros. These CMG's used two 107H size ball bearings with phenolic bearing retainers. The retainers were impregnated with KG-80 oil and served as a mechanism by which additional oil was centrifuged from the lube nut oil replacement system to the retainer flange and was then centrifuged through feed holes in the retainer to the I.D. of the bearing outer race.

A review of the bearing design parameters and further tests were instigated by the premature failure of CMG 1 (S/N 5) aboard Skylab, and anomalies encountered with CMG (S/N 6). These studies disclosed that in certain conditions most of the oil centrifuged from the retainer feed holes would not enter or migrate into the bearing outer race contact area (Ref. MT-15,776). Therefore, a redesign of the bearing retainer (Ref. MT-15,777) was performed such that oil from the flange would feed directly into the bearing outer race groove in all conditions. One set of holes would feed the oil into the area of the race groove above the ball

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path. The second set would feed oil directly into the center of the ball path under nominal conditions. In addition the retainer flanges were redesigned to further overlap the lube nut as well as to allow for the more acute angle of the oil feed holes.

3.0 DISCUSSION

To determine the effectiveness of the retainer redesign three life tests are being performed in life test fixtures and an additional life test is being performed in an ATM CMG Inner Gimbal and Rotor Assembly (IGRA). This report is concerned only with the three life tests performed in the life test fixtures.

The life test fixtures were designed originally to simulate to a certain extent the conditions of vacuum and weightlessness (lack of radial load) the bearing will encounter in an ATM CMG in space. The weight of the fixture shaft is negligible when compared with the 150 pound weight of the actual ATM CMG wheel. The life test fixtures are powered by an ATM CMG motor developing approximately nine oz-in of torque at 130 volts, 455 Hz, and at 9100 RPM shaft speed. The bearings are preloaded to 40 pounds, the same preload applied to the bearings used on the Skylab CMG's.

The life test fixtures were run in a spin axis vertical orientation such that the effect of gravity on the dispersion of the oil supplied to the bearings from the lube nut could be studied, as well as the effect of the pumping action of the retainer to disperse oil against gravity.

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To accomplish this, the test bearings in tests 1 and 2 were the bottom bearings in the fixtures and were minimally lubricated initially by being dipped into a 50-1 mixture of freon to KG-80 oil. The redesigned retainers in these test bearings were assembled dry except for a small quantity of oil in each ball pocket to prevent excessive friction between the balls and the pockets. Therefore, the location and amount of any oil in excess of the initial thin film on the metal parts would be directly attributed to the oil centrifuged from the lube nut to the retainer flange and dispersed by the retainer feed holes onto the outer race. Any further migration of the oil would then be the effects of migration due to gravity or pumping action of the retainer.

The test bearing in Test #1 was run with minimal lubrication and was resupplied with oil tinted with a red dye. The test bearing on Test #2 was similar to that in Test #1 except that the resupplied oil did not contain a dye. The life test bearings on Test #3 were wetted normally and contained normally lubricated bearing retainers as were the retainers on the bearings paired with the test bearings in tests 1 and 2. In each test the bearings were prepared as follows: the test bearing metal parts used in Test #1 were assembled for the first time specifically for this test and were prepared per (MT-13,923). The metal parts were then dipped into a 50-1 mixture of freon to KG-80 oil leaving a thin film of oil on the metal parts after evaporation of the freon. The redesigned retainers used in the bottom bearing (the test bearing) was soxhlet extracted in freon and baked dry at 220°F for 54 hours.

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Therefore, this retainer was completely void of oil except for the ball pockets of which each were "wetted" with approximately 1.5 mg of oil. This was accomplished by applying ten drops of a 50-1 mixture of freon to oil to each dry pocket with a #20 hyperdermic needle. This was done to reduce the friction and localized over heating in the ball pockets as would occur if the pockets were left dry. Therefore, the bottom bearing when assembled into the life test fixture contained only minimal oil to support an elasto-hydrodynamic film and to prevent rusting of the metal parts. Any additional oil found would necessarily have flowed from the lube nut/oil replacement system.

In Test #1 the top bearing used to pair with the test bearing was a bearing utilized in previous tests and did not contain a redesigned retainer.

The test (bottom) bearing in Test #2 and its redesigned retainer were prepared exactly as the test bearing and retainer in Test #1. In this test the oil in the lube nut to resupply the test bearing contained no red dye. In Test #2, the top bearing used to pair with the test bearing was also assembled for the first time for this test and the metal parts were cleaned per MT-13,923. However, the metal parts were dipped into the normal 5 to 1 freon to KG-80 oil mixture, leaving a thicker film on the metal parts. The retainer used in this bearing was a redesigned retainer and was fully impregnated.

In Test #3, both bearings were assembled for the first time for this test and the metal parts prepared per MT-13,923.

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The metal parts were then dipped into the 5-1 freon to KG-80 oil mixture to coat the metal surfaces as is normal for bearings for use on an IGRA. Likewise, the redesigned retainers used in these bearings were fully impregnated. The lube nuts used with these bearings also did not contain dyed oil.

4.0 RESULTS

As of December 6, 1974, Test #1 has run for 3588 hours and shows no visible signs of distress to the test bearing metal parts or retainer. There is a meniscus of oil visible between the balls and outer race contact surface. There is no evidence of degraded or pyrolyzed oil in the visible portions of the retainer pockets, on the retainer pads or in the bearing outer race. There are visible quantities of red tinted oil which have been pumped against gravity to the upward land of the outer race. There is excess oil in the outer race groove and contact area. There are greater quantities of red tinted oil which have migrated from the outer race groove downward with gravity to the bottom of the bearing (see Figure 4).

Visual examination of the bearing metal parts and retainers after 2101 hours in Test #2 and 3331 hours in Test #3 are similar to that in Test #1 in that there is only normal light wear banding visible on the bearing balls and outer race contact areas and no visible signs of metal or retainer distress. There is a visible meniscus of oil between the outer race contact area and the balls. There is likewise no visible evidence of any lubrication degradation. There is evidence of quantities of oil, pumped against gravity,

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visible on the upward land of the outer race of both bottom test bearings. There are larger quantities of oil visible in the race grooves and more which has migrated due to gravity from the race groove to the bottom of the lower land of the test bearings' outer races.

The progressive visible inspections of the lower bearing in Test #1 gives a graphic description of the flow and dispersion of the oil. At the 90 hour examination, see Figure 1, there is no visible additional oil on the outer race lands, and while there appears to be some additional oil in the outer race contact area and balls. The quantity is sufficient for a red tint to be noted. At the 328 hour examination (see figure 2), there is some evidence of additional quantities of oil on the bearing upper outer race land, pumped against gravity. There is definitely additional quantities of red tinted oil in the outer race land and balls, and a small meniscus of oil with a faint red tint between the balls and the contact area.

After the 1159 hour visual examination (see Figure 3), there is an additional quantity of red tinted oil pumped against gravity to the upward land of the outer race. The bearing outer race groove contains enough extra oil to be visibly pushed out of the ball contact area by the balls as the bearing is slowly rotated. There is a meniscus with a red tint visible between the balls and contact area. There is also additional oil from the outer race groove on the lower land and ending in a red ring at the bottom of the bearing. The examinations after the 2075, 2621 and 3588 periods (see Figure 4) show similar concentrations of red

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dyed oil in the races, balls, and upper land but with increasing quantities of oil being drained from the race groove as additional quantities are centrifuged into the bearing outer race.

The results of visual examinations of the bearings in Tests 2 and 3, are similar except that they are not as graphically visible since the oil used does not have a red tint.

Chart 1 below shows the hours and date of each visual examination of the test bearings.

Test/Visual Examination	1	2	3	4	5	6	7
Minimal Lube	90	328	1159	2075	2621	3588	
red dye	6/28/74	7/8/74	8/11/74	9/23/74	10/23/74	12/6/74	
Minimal lube	480	978	1677	2101			
(no red dye)	9/24/74	10/15/74	11/18/74	12/6/74			
Normal lube	484	1286	2384	3331			
(no red dye)	8/2/74	9/4/74	10/21/74	12/6/74			

CHART I

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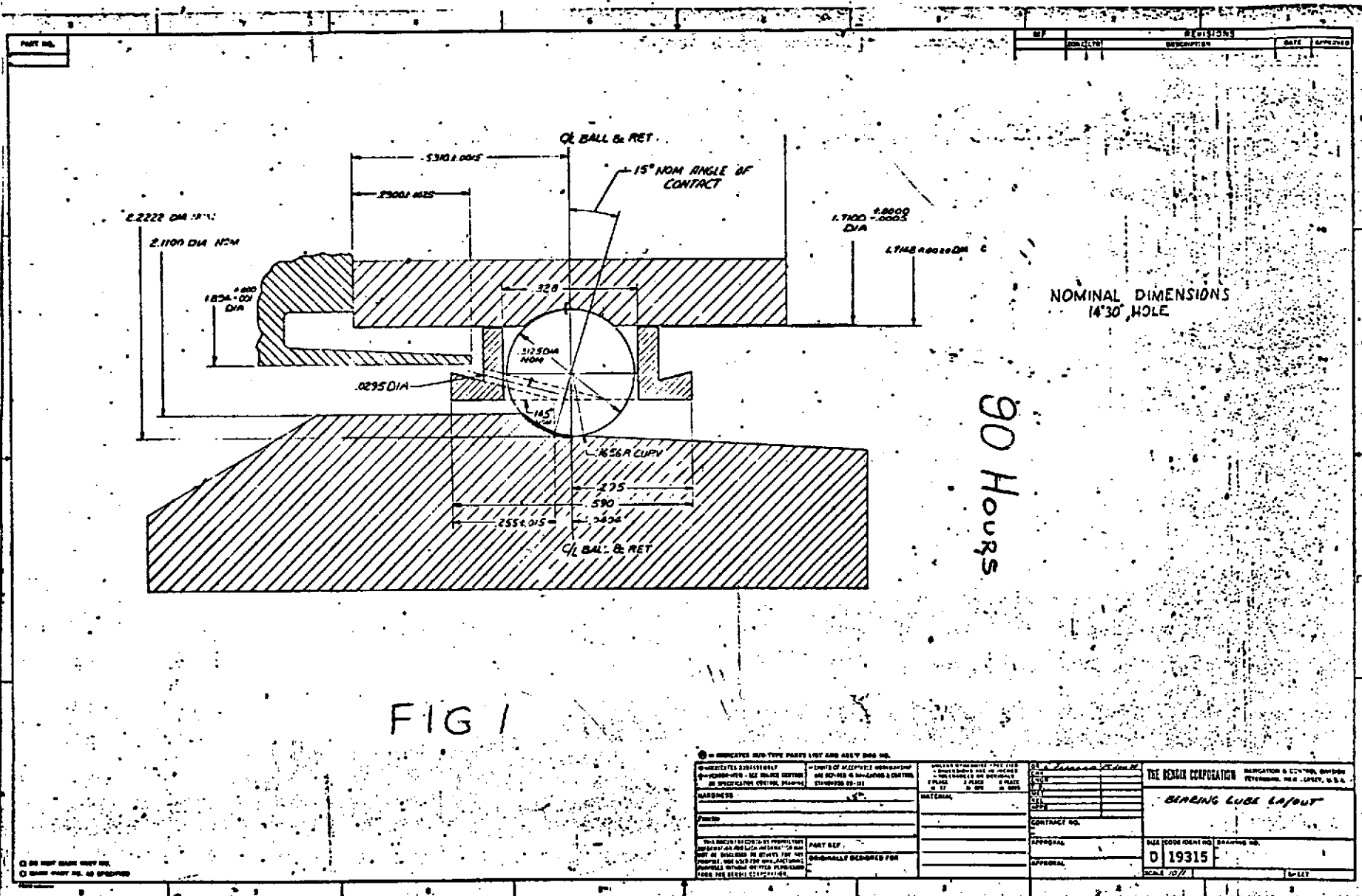
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5.0 CONCLUSIONS

The redesigned retainer does effectively capture all of the oil centrifuged from the lubrication replacement system and effectively distributes it to the bearing outer race groove and contact area.

The presence of oil on the upward land of the bottom test bearing in each test proves that some of the oil centrifuged into the bearing outer race is pumped upward against gravity through the race groove.

The fact that all of the oil centrifuged from the lube nut is captured by the retainer flange and is deposited into the outer race groove is proven by the absence of oil on the lower or lube nut side of the outer race land, during the first several hundred hours of each test. Only after sufficient excess quantities of oil are dispersed into the outer race does oil appear on the lower land due to drainage.



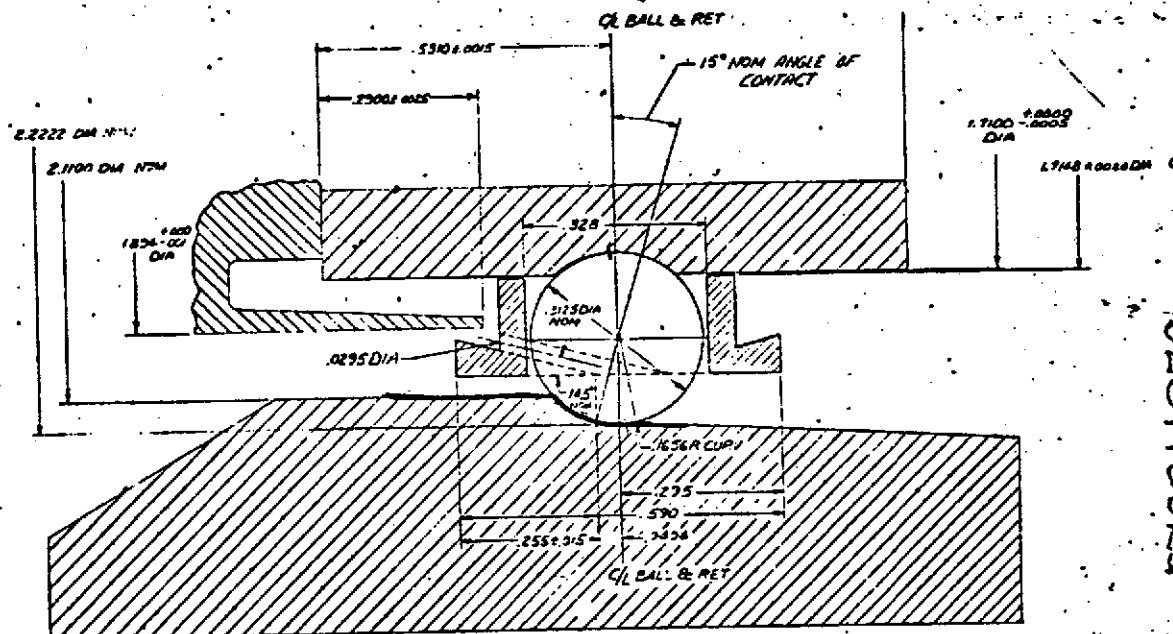
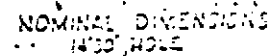


FIG 2

[illegible]



1159 Hours

FIG 3

[illegible]

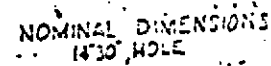


FIG 4

1. NAME (Last, First, Middle) 2. DATE OF BIRTH (MM/DD/YYYY) 3. PLACE OF BIRTH (City, State, Country) 4. CURRENT ADDRESS (Street, City, State, ZIP) 5. SOCIAL SECURITY NUMBER (XXX-XX-XXXX) 6. MARITAL STATUS (M, F, M, S, D) 7. OCCUPATION (Job Title) 8. EDUCATION (Degree) 9. MILITARY SERVICE (Branch, Grade, Dates) 10. OTHER INFORMATION (Notes)		11. PHOTOGRAPH (2x2 inch) 12. SIGNATURE (Handwritten) 13. DATE (MM/DD/YYYY) 14. OFFICIAL USE (Stamp) 15. COMMENTS (Handwritten)	
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To: Engineering File MT-15,781
From: C. Gurrisi

Issue: Original
Date: 10 December 1974

BEARING LIFE TEST WITH REDESIGNED
RETAINERS IN AN INNER GIMBAL AND
ROTOR ASSEMBLY (IGRA)

Prepared by: C. Gurrisi
C. Gurrisi

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY
07608

Issue: Original

MT-15,781

Date: 10 December 1974

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PURPOSE

This test program was initiated to validate and life test the present lubrication system with bearings containing a redesigned retainer in an inner gimbal and rotor assembly (IGRA). The main purpose was to demonstrate that the redesigned retainers would supply lubricant (KG-80 oil) directly to the ball/race contact zone while operating in a "1 G" field opposing the flow of the lubricant.

ABSTRACT

The failure of ATM-CMG No. 5 (Skylab CMG 1) and the anomalies which occurred on ATM-CMG No. 6 (Skylab CMG #2) resulted in an investigation of all aspects of the ATM CMG active lubrication system. As documented in MT-15,777, design changes to the ball bearing retainer were made to deposit lubricant directly into the ball/race contact zone. This test program was initiated to validate the design changes of the bearing retainer while operating in an IGRA with a one "G" field opposing the flow of lubricant to the race contact zone.

DISCUSSION

Inner Gimbal and Rotor Assembly S/N E-2 was assembled with two normally lubricated 107H angular contact ball bearings containing the redesigned retainers. After the unit was assembled an abbreviated functional test was performed to verify all performance parameters.

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This test accumulated a total of 36 hours on the bearings. Since all performance parameters were satisfactory the unit was partly disassembled for bearing visual inspection and installation of the active lubrication systems. The lubrication systems installed at this time had flow rates of .092 Mg/Hr on Side 1 and .073 Mg/Hr on Side 2. The unit was reassembled and a final acceptance and screening test was performed. This functional test is outlined below.

<u>Time</u>	<u>Operation</u>
0	1. Normal run-up to 8000 RPM (400 Hz, 115 Volts, Pump On)
24	2. Stable torque test spin axis horizontal, start-spin axis down
32	3. Stable torque test spin axis down, start stable spin axis horizontal
89	4. Stable torque test spin axis horizontal, start spin axis up
97	5. Stable torque test spin axis up, start stable speed spin axis horizontal
114:30	6. Stable torque test spin axis horizontal, run-up to 9100 RPM, 455 Hz, 130 volts
137	7. Stable torque test spin axis horizontal, start stable speed spin axis down
145:15	8. Stable torque test spin axis down, start stable speed spin axis horizontal
161	9. Stable torque test spin axis horizontal, start stable spin axis up

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<u>Time</u>	<u>Operation</u>
169	10. Stable torque test spin axis up, start stable spin axis horizontal
185:30	11. Stable torque test spin axis horizontal
257:30	12. Start off gassing tests
458:00	13. Stable torque spin axis horizontal, return to 8000 RPM
474	14. Stable torque test spin axis horizontal, start run down
479	15. End of test.

All performance parameters of this test are summarized in Table 1.

At the conclusion of this functional test the unit was vented, the side covers, end cap and strut assembly, and lubrication systems were removed to visually inspect the ball bearings.

It should be noted that the ball bearings were not removed, and the end shake (bearing preload) was not disturbed.

This inspection revealed that both bearing retainers were well lubricated. Both bearing inner raceways contained significant quantities of oil, and no evidence of polymerized, or degraded oil was present on the retainer pads. Although both retainers were wet with oil

PERFORMANCE PARAMETERS SUMMARY FOR 9-E-2 FROM 10/10/74 TO 10/30/74

SPIN AXIS HORIZONTAL	8000 RPM										9100 RPM										
1 BEARING & WINDAGE TORQUE (OZ)	3.80	4.83	3.53	3.69	3.46	3.73	3.86	4.30	4.43	6.60	8.03	8.76	8.30	8.52	8.76	6.23	5.90	6.03	5.76	5.06	4.33
2 BEARING TEMP SIDE 1 °F	86	85	88	88	85	89	83	91	89	89	91	91	91	91	95	72	92	93	93	95	86
3 BEARING TEMP SIDE 2 °F	83	87	85	86	83	85	85	86	95	95	94	96	97	97	99	97	96	97	97	95	88
4 BEARING VIBRATION SIDE 1 MV	18	11	12	15	14	27	22	21	28	27	27	27	90	90	90	28	28	28	30	30	29
5 BEARING VIBRATION SIDE 2 MV	13	11	13	15	14	24	25	20	25	26	26	27	50	50	50	28	28	29	30	29	28
6 CAVITY PRESSURE MICRONS	70	72	3.2	3.5	2.2	3.0	2.5	3.0	3.2	14.0	28.0	15.0	60.0	68.0	128.0	46.0	40.0	36.0	28.0	18.0	4.0
7 POWER WATTS	31	39	29	30	28	32	32	36	31	44	52	54	52	77	56	40	32	36	36	38	33
8 PEAK SPEED RPM	7914	7920	7943	7941	7942	9038	9036	9029	9034	9980	8942	8920	8909	8918	8849	8975	8990	8994	9000	9016	9030
9 HOURS SPIN AXIS HORIZ.	24:00	27:30	17:20	28:20	30:00	23:40	15:30	16:30	88:00	112:30	136:00	160:00	184:00	171:30	258:00	257:00	260:00	262:00	263:00	264:30	281:00

SPIN AXIS UP

1 BEARING & WINDAGE TORQUE (OZ)	5.66				6.13
2 BEARING TEMP SIDE 1 °F	88				89
3 BEARING TEMP SIDE 2 °F	90				92
4 BEARING VIBRATION SIDE 1 MV	11				31
5 BEARING VIBRATION SIDE 2 MV	21				32
6 CAVITY PRESSURE MICRONS	4.2				3.2
7 POWER WATTS	40				44
8 PEAK SPEED RPM	7895				9185
9 HOURS SPIN AXIS UP	8:30				8:00

GENERAL DATA

1. RUN-UP TIME HOURS (7800RPM 400HZ 115 VOLTS 30 POWER SUPPLY)
2. AXIAL END SHAKE (INCHES)
3. SPEED PICK-UP GAP (INCHES)
4. RUN TIME OF TEST (HOURS)
5. BEARING TEMPERATURE AT START OF

SIDE 1
SIDE 2

9 Hr.
0.0119
0.0095
480 Hz
70°F
73°F

SPIN AXIS DOWN

1 BEARING & WINDAGE TORQUE (OZ)	5.63				5.93
2 BEARING TEMP SIDE 1 °F	92				96
3 BEARING TEMP SIDE 2 °F	85				86
4 BEARING VIBRATION SIDE 1 MV	12				21
5 BEARING VIBRATION SIDE 2 MV	11				22
6 CAVITY PRESSURE MICRONS	6.5				3.5
7 POWER WATTS	38				43
8 PEAK SPEED RPM	7883				8991
9 HOURS SPIN AXIS DOWN	8:00				8:00

TABLE 1

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there were no large accumulations of oil in the retainer edge undercut as found in some of the bearings life tested with the original retainer design. Therefore the design changes of the oil feedthrough holes provides more efficient oil flow directly into the ball/race contact zone.

The unit was reassembled and accelerated to 9100 RPM in the spin axis horizontal position, this unit is currently on life test in the spin axis down position at BGSD. To date this unit has accumulated 167 hours in this orientation. When 1000 hours of operation in this attitude are accumulated, a bearing visual inspection will be performed, and if the results are satisfactory, the test will continue.

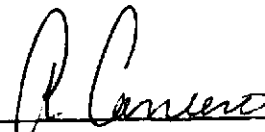
To: Engineering File MT-15,777
From: A. Carriero

Issue: Original
Date: 24 October 1974

ATM CMG BALL BEARING

RETAINER REDESIGN

Prepared by:


A. Carriero

THE BENDIX CORPORATION
GUIDANCE SYSTEMS DIVISION
TETERBORO, NEW JERSEY

07608

Issue: Original
Date: 24 October 1974

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Page 1

1.0 PURPOSE

The purpose of this report is to document the modifications made to the ATM CMG Ball Bearing Retainer. These changes are required to more effectively distribute the oil from the annular groove on the side of the retainer to the bearing outer race.

2.0 BACKGROUND

A complete design review of the ATM CMG system was required due to the failure of CMG S/N 5 on Skylab and the speed changes in CMG S/N 6 also on Skylab. During review of the ball bearing phenolic retainer it was noted that although the center of the oil feed hole was well past the edge of the outer race dam, the rear of the hole, where the oil drops actually formed, did not clear the edge. Therefore, the oil drops centrifuged from the rear of the feed hole, onto the outer race land approximately .027" from the dam edge, and migrated into the outer race. There is a question as to whether sufficient oil migrated into the outer races of the CMG's on Skylab and affected performance or propagated the problems encountered in CMG S/N's 5 and 6. The third CMG, S/N 7 ran without any problems noted, throughout the entire 6,500 hour mission in space.

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3.0 DISCUSSION

The redesign of the retainer was effected to assure that oil droplets would centrifuge directly into the outer race in all cases. To accomplish this the following changes were made:

- 1) The original holes were drilled at a more acute angle to overlap the race dam by a minimum of .018" in the worst conditions and .0315" at nominal conditions. See Figures 1, 2 and 3.
- 2) An additional set of holes were added to centrifuge oil droplets into the bearing outer race at the nominal contact angle of 14.5° , see Figure 4.
- 3) The width of the oil groove was increased to allow for the more acute angle of the oil holes, without loss of groove dam diameter. The additional width also increases the overlap of the groove with respect to the lube nut by .036" from a minimum of .0035 to .0395. See Figures 5 and 6.
- 4) The retainer outer diameter was increased by .012". This decreases the minimum radial clearance between the outer race land and retainer O.D. from .026" to .020". However, it allows the nominal 19° angle hole to extend .017" further into the outer race, and the nominal 14.5° hold .023" further extension into the outer race.

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The oil droplets will now centrifuge off into the bearing outer race in two areas: One area is above the contact zone from which oil will migrate into the contact zone of the outer race. The second area is in the center of the nominal contact zone. Therefore, the bearing outer race contact zone will be lubricated both directly and indirectly providing an adequate supply of oil to be transferred by the ball to the inner race and retainer pocket and ensuring an adequate hydrodynamic oil film. See Figure 7.

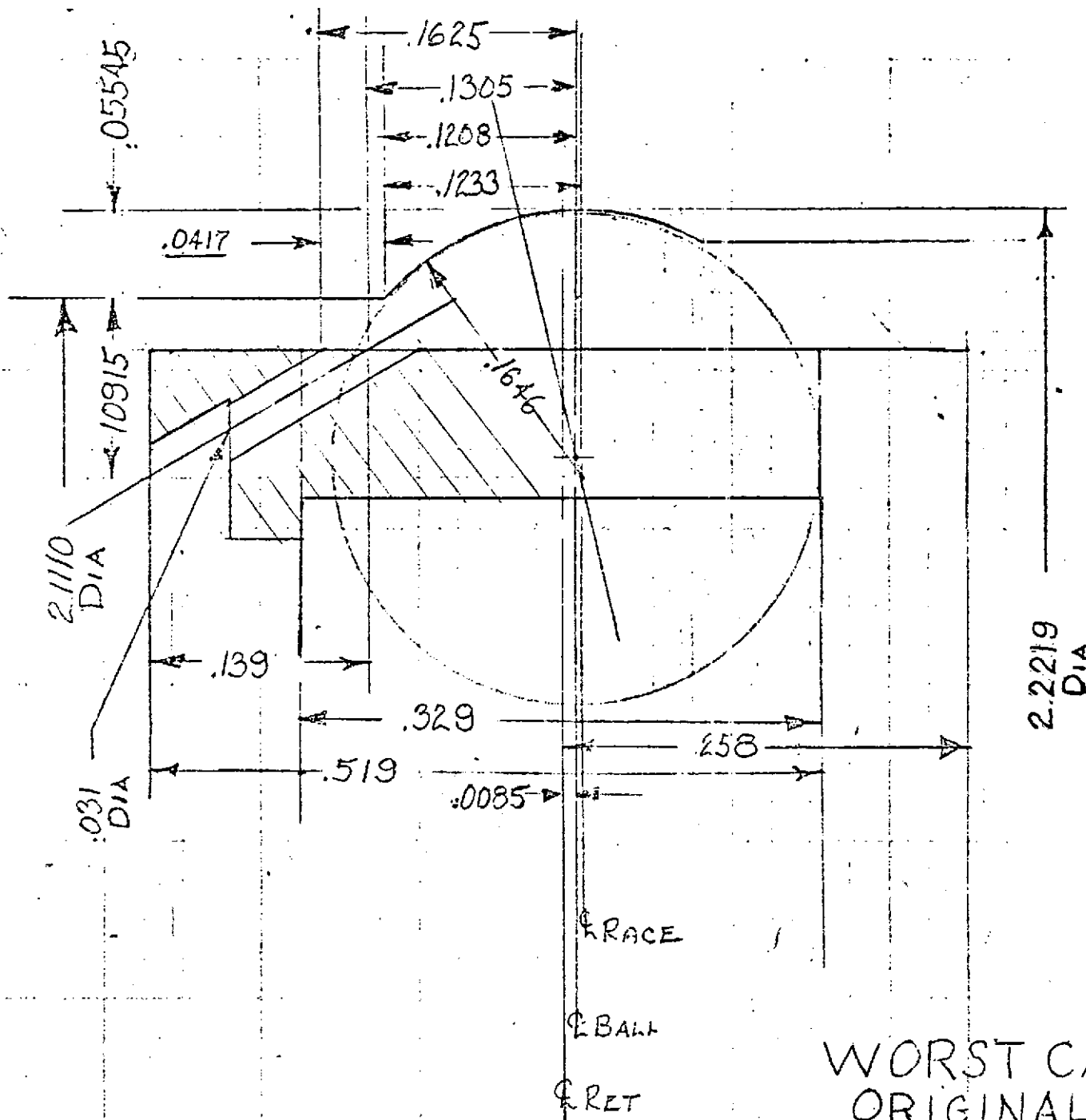
RETAINER DESIGN CALCULATIONS

	WORST CASE 29° HOLE-FIGURE 1	WORST CASE 20° HOLE-FIGURE 2	NOMINAL CASE 19° HOLE-FIGURE 3	NOMINAL CASE 14.5° HOLE-FIGURE 4
A. CONTACT ANGLE ϕ	17°	17°	15°	15°
B. BALL SIZE	.3120	.3120	.3125	.3125
C. BALL POCKET DIA.	.329	.329	.328	.328
D. OUTER RACE CURVATURE	.1646	.1646	.1656	.1656
E. OUTER RACE GROOVE DIA.	2.2219	2.2219	2.2222	2.2222
F. OUTER RACE LAND DIA.	2.1110	2.1110	2.1100	2.1100
G. OUTER RACE GROOVE DEPTH (E-F)/2	.05545	.05545	.05610	.05610
H. OUTER RACE CENTER TO LAND CORNER	$.1233 - [(1646)^2 - (.1646 - .05545)^2]^{1/2}$		$.1242 - [(1656)^2 - (.1656 - .05610)^2]^{1/2}$	
I. BALL CENTER TO LAND CORNER	$.1208 = .1233 - (.1646 - .1560) \sin 17^\circ$		$.1218 = .1242 - (.1656 - .15625) \sin 15^\circ$	
J. RETAINER DISPLACEMENT	.0085	.0085	0.000	0.000
K. BALL CENTER TO OIL HOLE LIP	$.0085 - .258 + .519$ $-.139 + (.031/2) \sin 29^\circ$ $= .1625$	$.0085 - .294 + .591$ $-.203 = .1025$	$.295 - (.250 - \frac{.01475}{\sin 19^\circ})$ $= .0903$	$.295 - .255 = .040$
L. OVERLAP - OIL HOLE LIP TO LAND CORNER	$-.0417 = .1208 - .1625$	$+.0183 = .1208 - .1025$	$+.0315 = .1218 - .0903$	$+.0818 = .1218 - .0400$
I. LOCATION - BALL CENTER LUBE NUT	$.5310 + .0015 = .5325$ $.2900 - .0025 = .2875, .5325 - .2875 = .2450$			
II. LOCATION - BALL CENTER TO RETAINER GROOVE	$.517 - .260 - .0085$ $= .2485$	$.589 - .296 - .0085 =$ $.2845$		
III. OVERLAP - RETAINER GROOVE TO LUBE NUT	$.2485 - .2450 = .0035$ SEE FIGURE 5	$.2945 - .2450 = .0395$ SEE FIGURE 6		

TABLE I

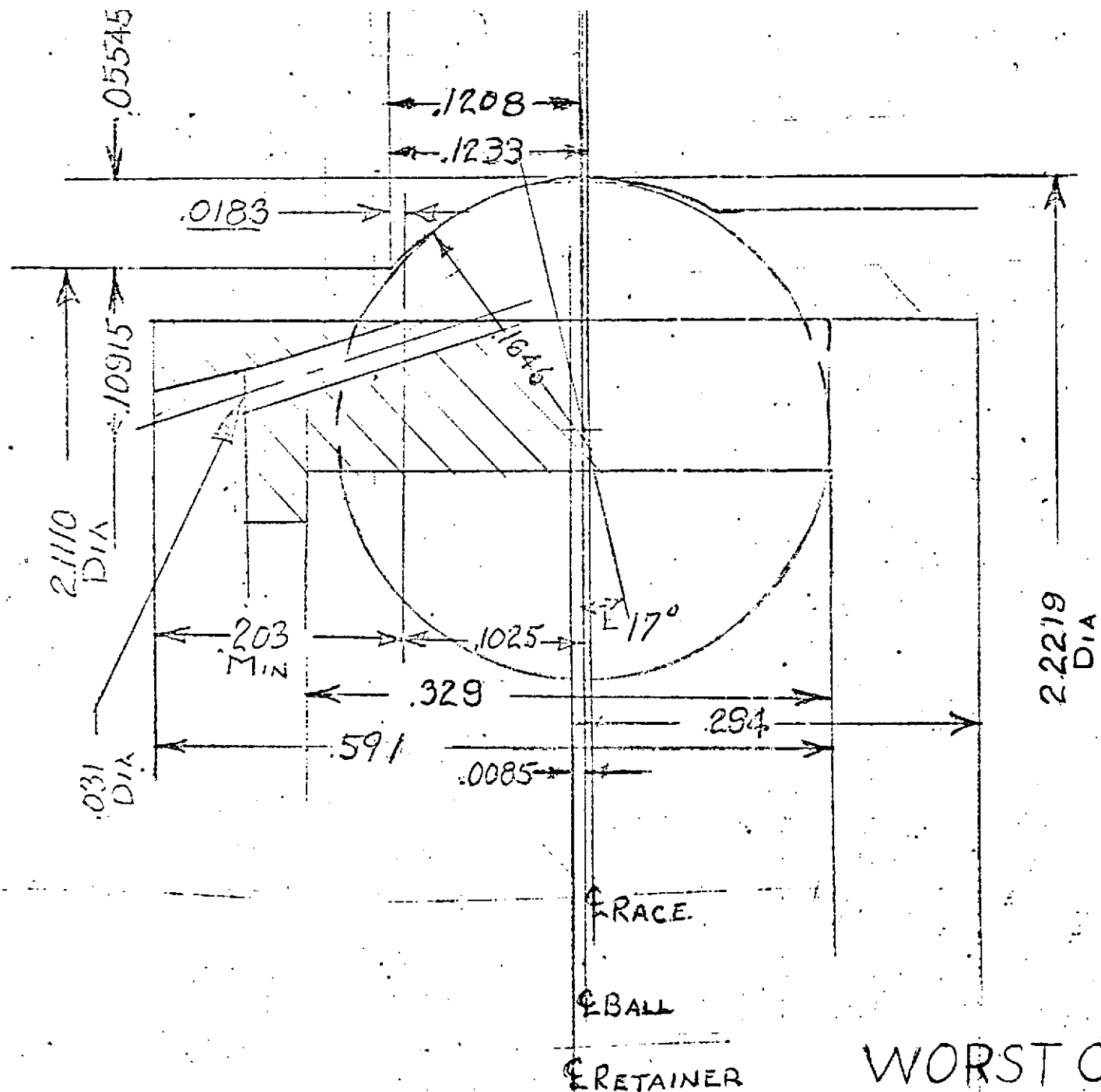
Issue: Original
Date: 24 October 1974

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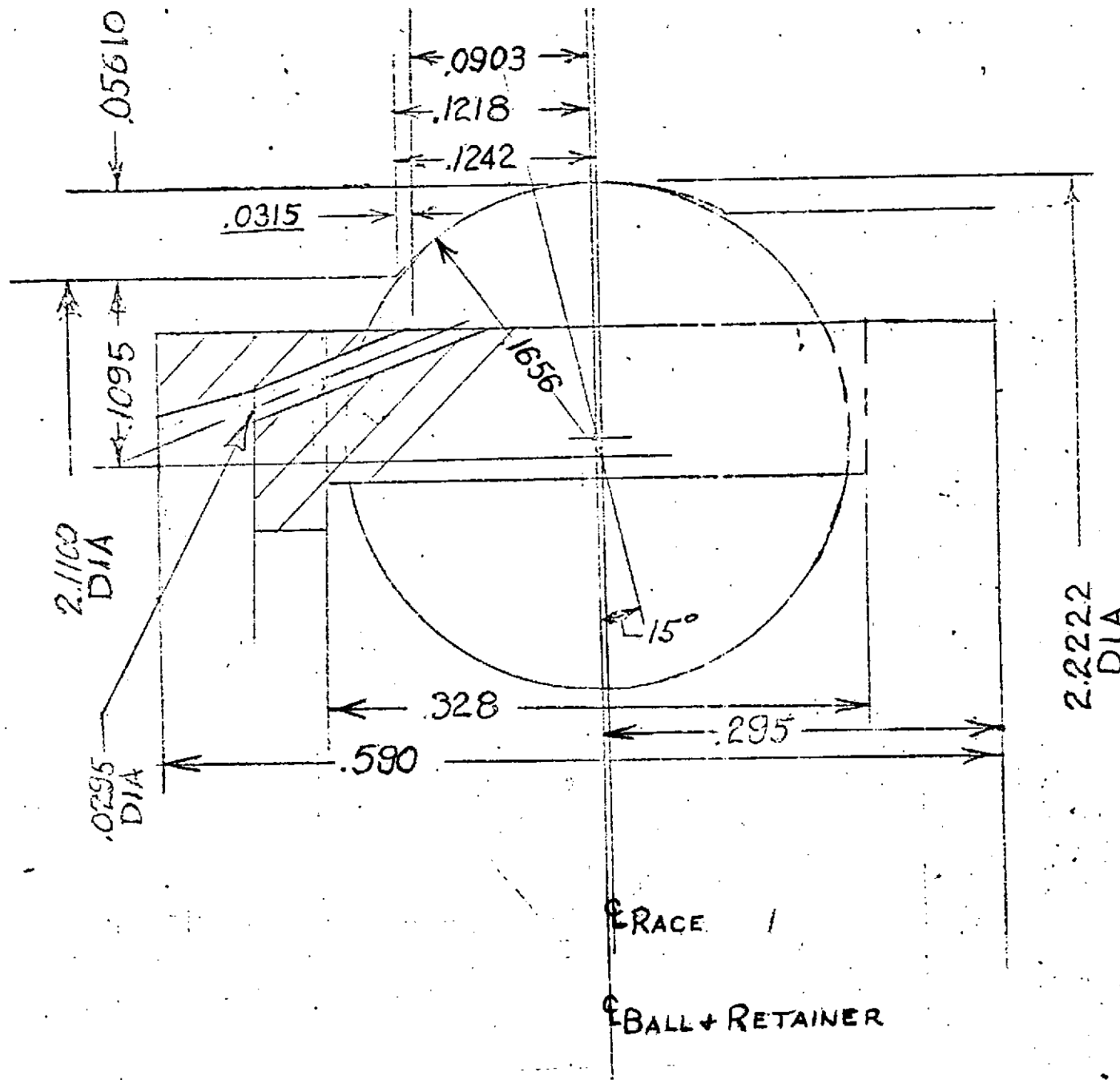
WORST CASE 29° HOLE
ORIGINAL RETAINER

FIGURE 1



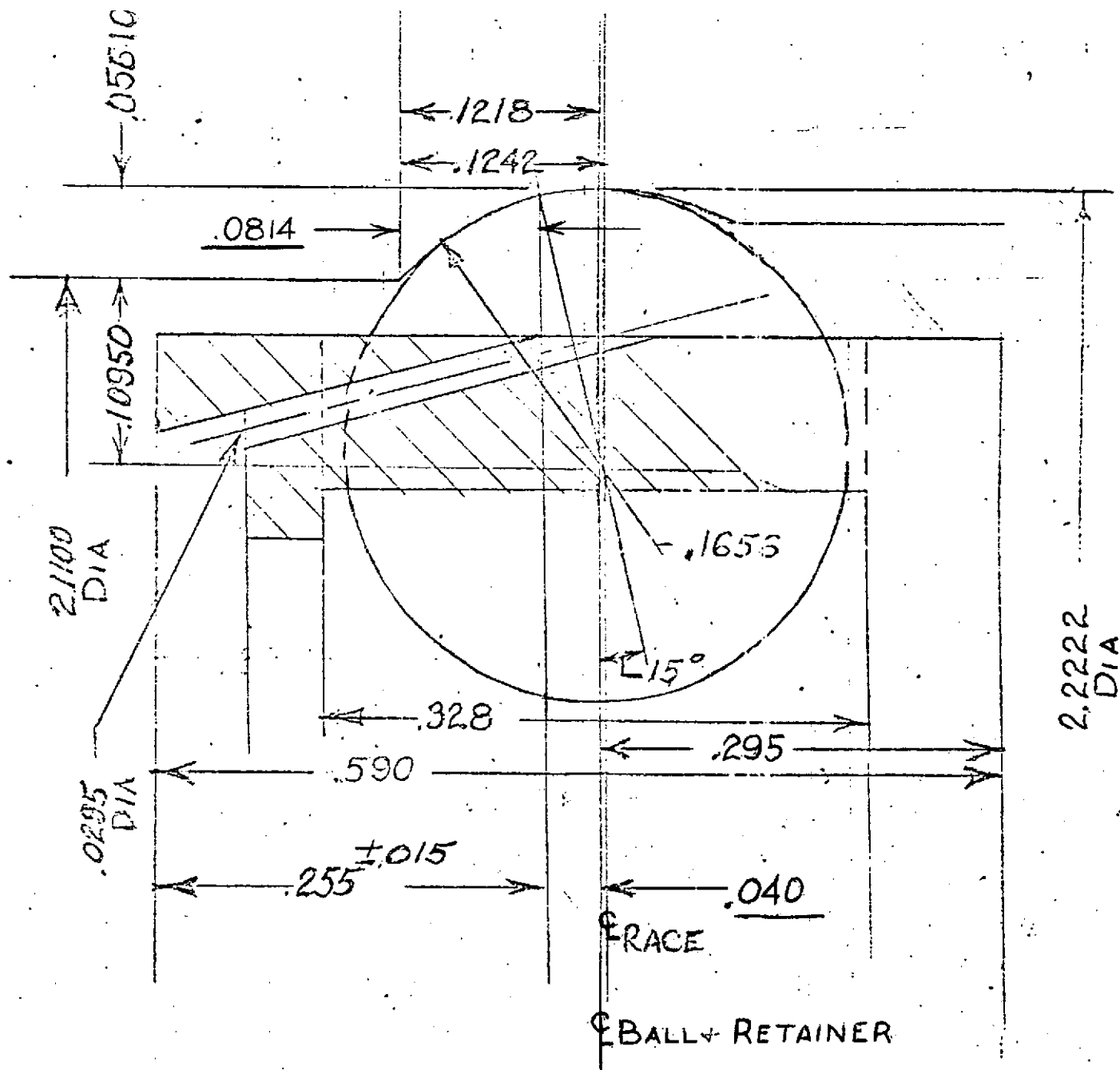
WORST CASE 20° HOLE
REDESIGNED RETAINER

FIGURE 2



NOMINAL CASE 19.0° HOLE

FIGURE 3



NOMINAL CASE 14.5° HOLE

FIGURE 4

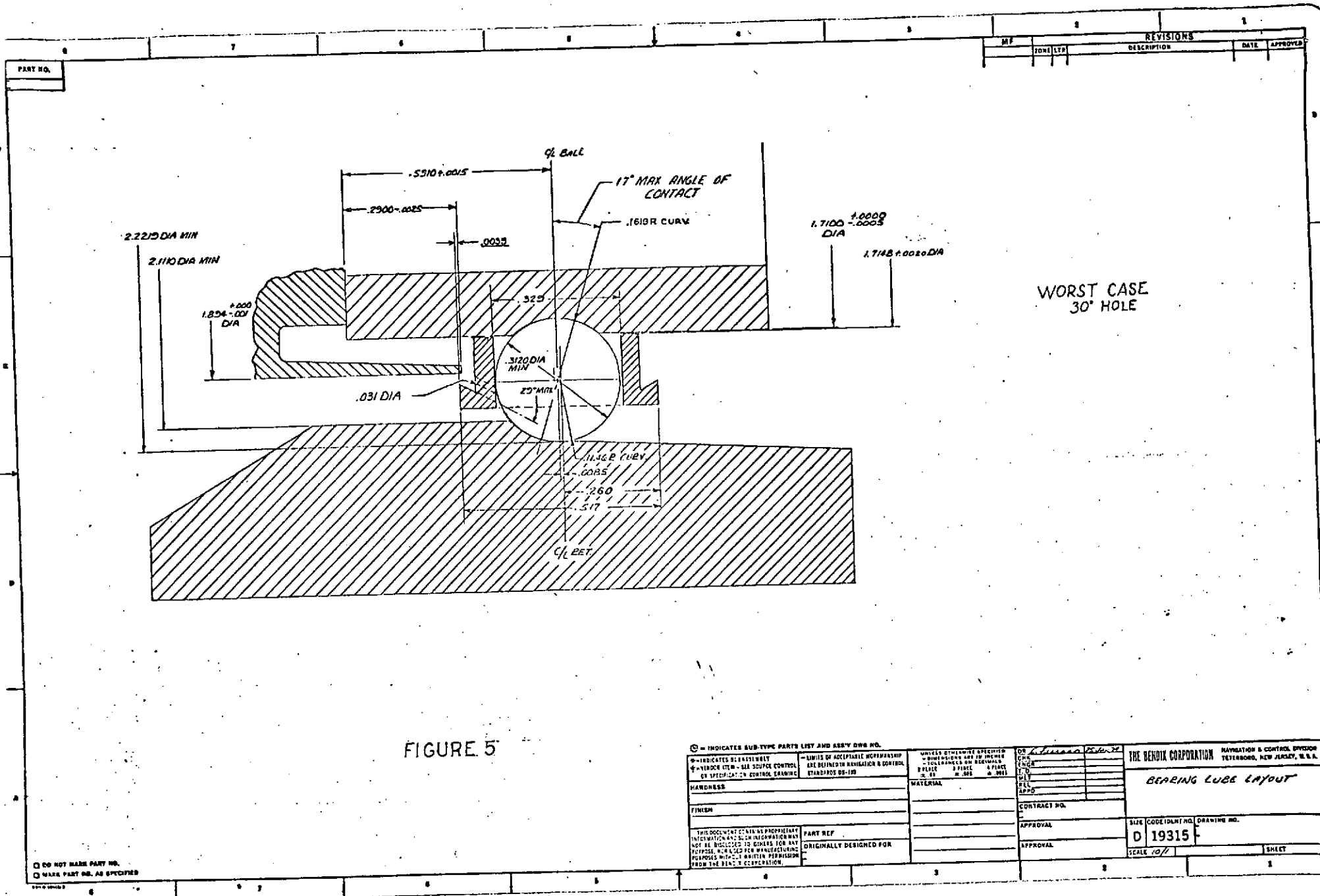


FIGURE 5

DO NOT NAME PART NO.
NAME PART NO. AS SPECIFIED

@ INDICATES SUB-TYPE PARTS LIST AND ASSY DWG NO. @ INDICATES ELASTICITY @ INDICATES ITEM - SEE SOURCE CONTROL OR SPECIFICATION CONTROL DRAWING		LIMITS OF ACCEPTABLE WORKMANSHIP ARE DEFINED IN NAVIGATION & CONTROL STANDARDS DS-100		UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES ON DECIMALS .010 .005 .001		ON CHG CORR ETC APPD		THE BENDIX CORPORATION NAVIGATION & CONTROL DIVISION TETERBORG, NEW JERSEY, U.S.A.	
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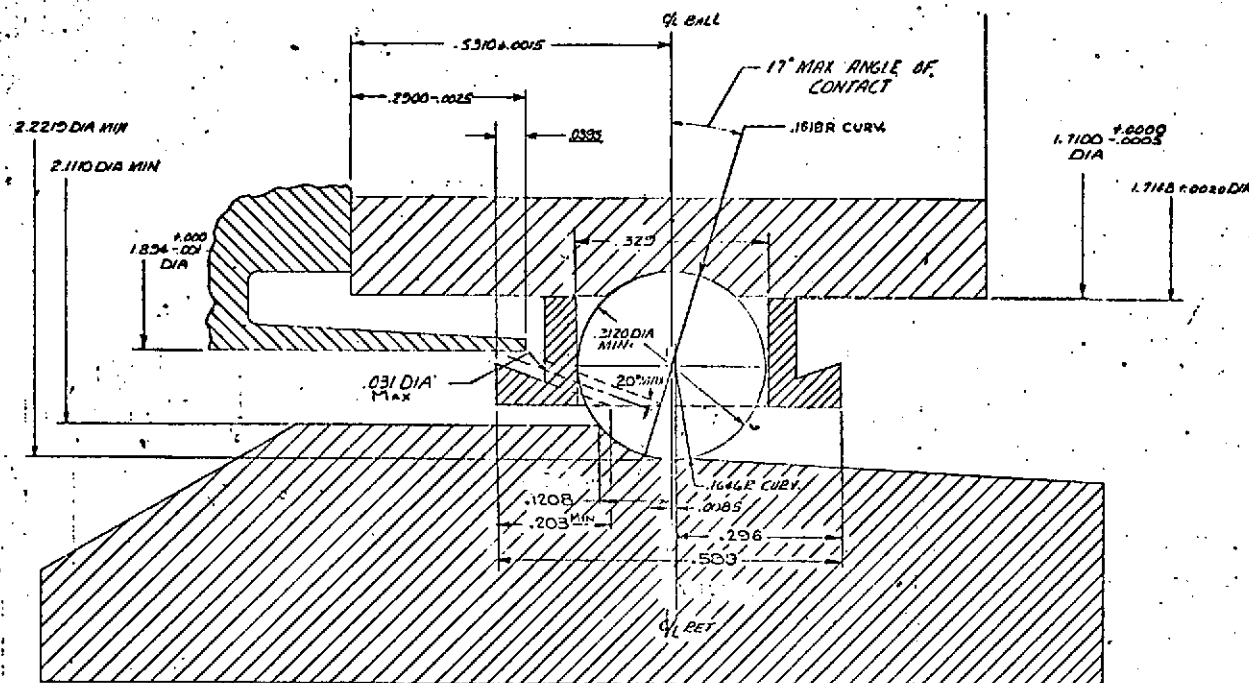
ZONE LTR

REVISIONS

DESCRIPTION

DATE

APPROVED



WORST CASE
20° HOLE

FIGURE 6

DO NOT MARK PART NO.
MARK PART NO. AS SPECIFIED

⑤ INDICATES SUB-TYPE, PARTS LIST AND ASS'Y DWG. NO.

R-INDICATES SUBASSEMBLY
+TOLERANCE FILM - SEE SOURCE CONTROL
OF SPECIFICATION CONTROL DRAWING

± INDICATES SUBTYPING
ARE DEFINED IN SPECIFICATION & CONTROL
STANDARDS 05-100

WHILE OTHERS ARE SPECIFIED
- TOLERANCES ARE IN INCHES
- TOLERANCES ON DECIMALS
P. 01 P. 02 P. 03 P. 04 P. 05

CR-INDICATES SUBTYPING
- TOLERANCES ARE IN INCHES
- TOLERANCES ON DECIMALS
P. 01 P. 02 P. 03 P. 04 P. 05

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SIZE CODE IDENT NO. DRAWING NO.
D 19315

SCALE 1/8" = 1" SHEET

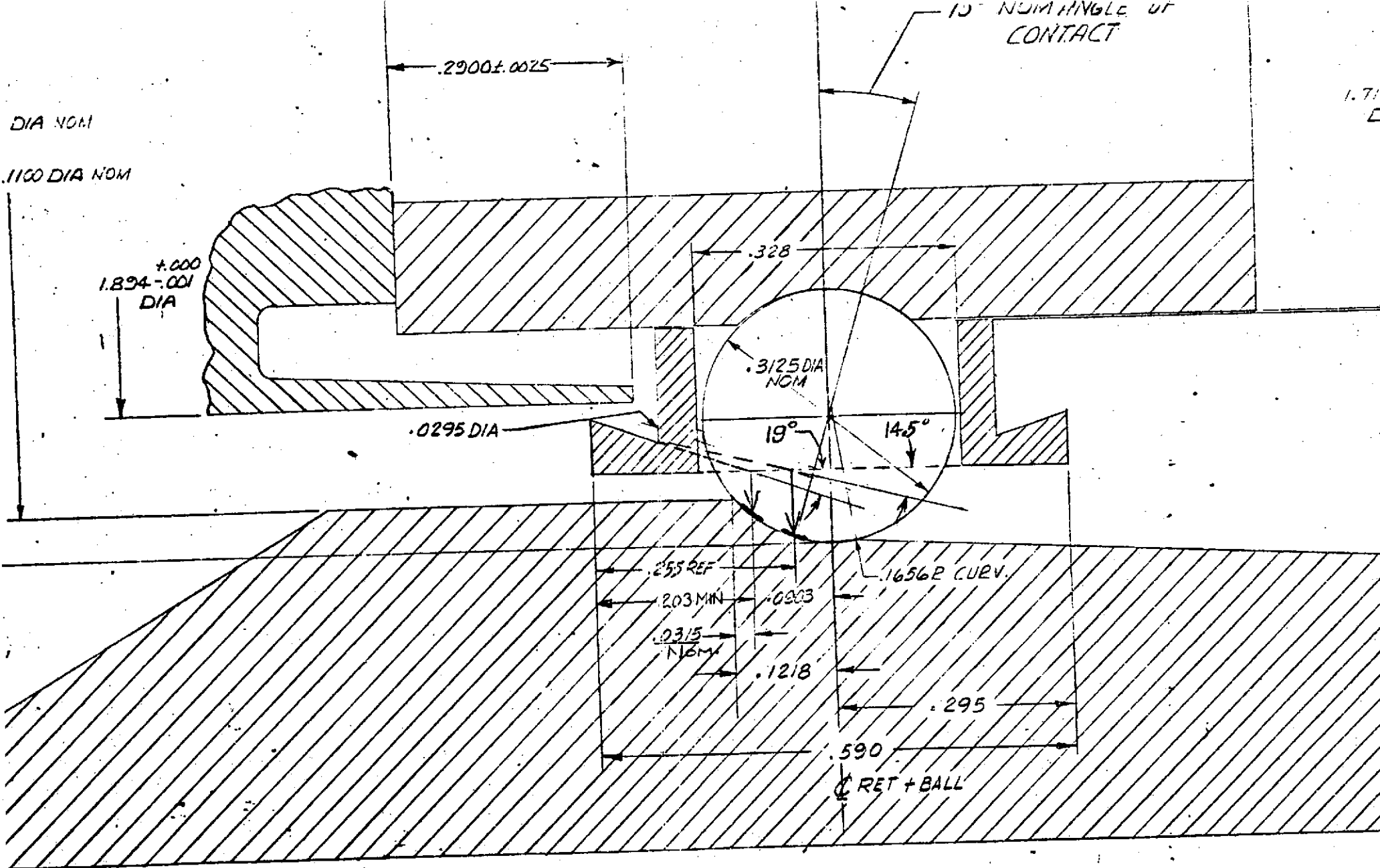


FIGURE 7
OIL DEPOSITION